COMPRESSED AIR PRACTICE

Electric Air Drill in a Granite Quarry. *Frontispiece*

COMPRESSED AIR PRACTICE

BY

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BEFORE BEGINNING

The material comprising my little book, "Compressed Air" some small portion of which may be recoginzed in the present volume, was written twenty years ago. In the two decades which have elapsed there has been a great advance in the industrial status of compressed air, and a wonderful development and extension of compressed air practice, with many radical changes, and the writer in the meantime has had to unlearn perhaps as much as he has learned.

It is only the simple truth that the average power and fuel cost of the compressed air now used—compressed by steam power, directly or indirectly, as most of it still is—is not more than one-third as great, per quantity unit, as it was twenty years ago; and perhaps there has been, but concerning this no assertion is here made, an equal augmentation of advantage in the useful applications of the air. As to the increase in the twenty years in the volume of air more or less compressed and industrially employed a thousand fold may be a conservative guess, while to say a hundred fold would suggest an excess of caution or a deficiency of information.

It follows that with the vast extension of the field of compressed air employment, there has been a corresponding accumulation of knowledge concerning it, an increase in the number and variety of the things to be told about it, leaving the scope of the inquiry as to the possibilities beyond the present more extensive than ever. This book just tells what it can, within the permissible limits of it, leaving still more than ever unsaid, but perhaps suggesting things to be supplied by others who may come after.

We designedly have to do here with the general rather than with the specific, with the approximate rather than the precise. The book is intended for the many rather than for the few, for those who know little about air rather than for those who mostly know all that is known. Those who need more accurate and detailed information for practical guidance may perhaps get some hints here as to the how, if not so much the where, to look for it.

There has developed within a very few recent years a special line of publications put forth by the builders of air compressors and by the manufacturers of the various air actuated tools and apparatus. These publications, usually costing the reader nothing, well written, excellently illustrated and printed, and easy to handle, embody a great variety and reach of definite and generally reliable information which it is not the purpose of this book to reproduce in any form. Catalog "literature" and auctioneer talk have their place, but not here.

Frank Richards.

New York,
November, 1913.

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COMPRESSED AIR PRACTICE

CHAPTER I

ATMOSPHERIC GENERALITIES

As all the air that we come in touch with, or about which we know anything, is compressed air, less compressed upon the mountain tops, or in the spaces that the aviators tumble from, and more compressed in the lowest deeps of caves and mines by the weight of the air always above it; if we speak of air at all, and without any specification as to its condition, we necessarily must be speaking of compressed air, and in going through the pages which follow it may be well to keep the fact in mind, instead of having to remind ourselves of it every time that air or *the* air is mentioned.

It is not easy to recognize or to appreciate all that the air, compressed air if you please, is to the earth as a whole, or to comprehend the many and diverse things it is employed to do, and which it alone can do, in the active workings of nature. A partial running over in our minds of some of its world functions, not knowing even yet what they all are, nor any of them with any completeness, should crowd our minds with suggestions of ways in which we may employ it to advantage, and may lead us to regret and wonder that we have not already more fully availed ourselves of its ready ability to serve us.

The total weight of the earth's atmosphere is, say, only one-millionth of the weight of the earth as a whole; yet that one-millionth would seem to be of more value and importance than any other specific constituent, however vast, of the entire mass. The total volume of the waters of the earth weighs perhaps 300 times as much as the air, yet the air is still the more important and necessary, if, indeed, such a comparison is permissible where both are indispensable.

In the artificial development of power from the active forces of nature the windmill as such makes but a pitiful showing when compared with the might of the constantly growing host of water-turned wheels; yet Niagara and Victoria and all the other waterfalls, little and big, and the rivers and flowing streamlets everywhere, and all the tossing waves which make playthings of the ships, are actuated ultimately and entirely by power of which the air is the vehicle, and every waterwheel is a windmill in disguise. The waters are all lifted from their ocean reservoirs by the atmosphere, and are carried by it and delivered to flow all over the land and through all the valleys down to the sea, and then the air draws them up and carries and distributes them again; and so evermore the air drives all the natural and all the artificial activities of the world. The air is the breath of life of the earth, as it is of all who dwell upon it.

Man is constantly and absolutely dependent upon the air for his life and for all satisfaction and accomplishment in life. He can survive without solid food three weeks, without water three days, but without air for only three minutes. Of his catalogued senses those of hearing and smell are directly air actuated. The eye receives its most vivid and memorable impressions from the constantly recurring air-born phenomena of the skies, the changing draperies of the clouds all day, the gorgeous climax of the sunset and the sombre draperies of the night; water can act directly upon the touch alone.

Air is the breath not only of man and the land animals but also of the plants, the abstracting from the atmosphere of certain elements by the one balancing and compensating for that taken by the other, both in the process exchanging what they must discard for what they most need. By air pressure necessary gases are retained in water, in the blood of animals and in the sap of plants.

The air, we learn, is merely a mechanical mixture and not a chemical combination of gases, one or the other of its principal constituents being abstracted in varying quantities in sustaining the functions of universal life, and yet the proportions of the mixture are automatically maintained by the balanced operations of nature, while to man with a full supply of ingredients and apparatus it would be impossible to produce a single breath of "fresh" air.

While the air might, in a way, be called a jack-of-all-trades, it does not work by makeshifts, but is perfectly equipped for its several functions. Perhaps first in the kit which it carries is aqueous vapor, and this it keeps constantly active, the meteor-

Courtesy of Sullivan Machinery Company.

Fig. 1.—Undercutting in Mine of Cowanshannock Coal & Coke Company, Yatesboro, Pennsylvania.

ological reports showing the fluctuations of atmospheric humidity to have a width of range and a rapidity of change with which none of the other reported phenomena can compare; but then the normal air carries also carbon dioxide, ammonia, ozone, acid compounds of nitrogen and sulphur, and small samples of many other gases, with a miscellaneous collection of dust particles and of germs and seeds innumerable, all of which are heard from in their turn.

While we all of us must have some respect for the atmosphere, and while we may understand in a general way that we couldn't get along without it, still many of us have an idea that it does its work in a rough, or as we might say, in a careless or unprecise way, notwithstanding that it "gets there" every time; but, as man seems to be slowly finding out, some of the littlest work as well as the biggest is done by the air. The swift and untiring flight of the bird of broad and sweeping wing is impressive, but we may not forget that the air is also the free highway for myriads of minute organisms down to invisibility and far beyond.

The functions of the air would seem, from the immediate human viewpoint, to be not all nor always beneficent. Not only are we dependent upon it for life and health, but it also carries disease and, it may be, death, so that now and always the air is the most important single matter to which healers and sanitarians can turn their attention. This in actual life would seem to be little realized by us, since it is so notorious that we habitually are much less fastidious and exacting about the composition and condition of the air we breathe than we are about our food and drink. Air is always ready to provide the active agencies of decay, so that the function of excluding the air from edible materials which are to be kept for a length of time is the foundation of some of our most important industries.

The air does not always work in the mass, but is employed by animated selective agencies in minute isolated portions for delicate and responsible operations. Let us remember the air bladder of the little fish. What more wonderful than the perfect poise of a fish in the water? yet the device by which principally it is maintained is of the simplest character, but no human ingenuity can suggest how the air function could be dispensed with.

The most alert and the swiftest fish of prey with which we are familiar, such, for instance, as the pike or the pickerel, seem to spend much of their time absolutely without change of position, watching and waiting in readiness to dash at their prey, as upon land the cat watches the bird or the mouse.

A submerged body charged with life forces, but for the time floating inert in the water, is apparently free to move, or rather to be moved, in any direction, and that it should not so move is probably the most unlikely thing which could be predicted of it, yet fish seem to be provided with the means of resisting or of counteracting the minutest as well as the strongest of disturbing forces, and to be able to maintain a position when so desired apparently without effort, and in fact almost automatically, so that they often seem to sleep in a running stream as though anchored there.

To retain its position vertically, so that it will neither rise nor sink in the water, the body of the fish as a whole must have precisely the same weight as the volume of water which it displaces, and means must be provided, and they are so provided, for the instant adjustment of the specific gravity of the fish body. The air bladder of the fish, normally filled with compressed air, is a device for the purpose which for simplicity, precision and effectiveness is perfection.

The use of compressed air in the bladder of the fish is perhaps the most pronounced instance we have of "compressed air" practice entirely unassociated with human agencies. The air in the fish bladder must be always under more or less pressure, such pressure as would be easily measurable and recordable by our gages if only it were accessible. The pressure, indeed, may be considerable, as we are only familiar with the bladder after removal from the dead fish, and which although still distended by sensible air pressure is released from the muscular compression which is normal to it in the living fish. The pressure of the air in the bladder opposes a constantly maintained muscular tension in the body which encloses it. When the muscular tension is increased the air is still more compressed, giving the bladder a diminished volume, and the entire body of the fish in which the bladder is enclosed is correspondingly reduced in bulk, so that, while its actual weight remains constant, its weight relatively to the weight of the water displaced is increased, and the tendency of the fish body is then to sink in the water.

On the other hand, if the muscles which hold the air bladder and its contents in compression are relaxed, the bladder, and then the body of the fish, will be distended and the fish will have a tendency to rise in the water. When fish are not well and have not the muscular strength to maintain the requisite compression they float to the surface. If a mermaid belle in a ballroom of the deep should faint away she would not fall to the floor, but would float to the ceiling and become entangled among the chandeliers.

The air bladder of the fish has a still more delicate and responsible function than that of merely maintaining the precise specific gravity of the body. The stability of the fish laterally in the water is assured by the fact that the air bladder is normally located high in the body and the preponderance of weight is below it. The longitudinal, or fore-and-aft, trim of the fish is more difficult to adjust and maintain, as builders and operators of submarine craft have learned. This also the air bladder perfectly provides for, it being of considerable length relatively to its diameter and also partitioned transversely or divided into two chambers, the muscular control of the fish transferring air from one chamber to the other as may be required. The fish floating in liquid of uniform weight and density is thus perfectly equipped, with slight local adjustments by the aid of its fins, for maintaining its position and balance with ease and accuracy.

The atmosphere is one of the instruments by whose employment nature does some of its finest work, and, relatively to earth as a whole, its functions are adjusted with micrometric precision. A favorite plaything of the writer—in his mind—is an 8-in. terrestrial globe. The precise size of it is important here as it will enable us to do our thinking to scale, a very necessary particular, here and elsewhere, if our thinking is to be of any account. Earth is understood to be about 8,000 miles in diameter, and on our 8-in. globe, therefore, an inch will represent a thousand miles and a thousandth of an inch will represent a mile. This at once makes things on earth's surface look very small, and the heights and the depths, which make our mountain and valley scenery for globe trotters to patronize, become too minute even for detection.

If there were a fellow big enough to hold earth in his hand like a baseball, its surface would, on the same scale, appear to him smoother and more highly polished than glass. If we assume our 8-in. globe to be as smoothly finished as possible all over its surface, and if we paste upon it a bit of paper a thousandth of an inch thick, the thinnest tissue paper we can get, that will of course be a mile high, and it will represent the range of altitude within

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which 95 per cent. and more of the human race live and move and have their being. Dwellers upon the plains and the prairies, who know nothing in their actual lives higher than a two story or a three story house, may be assumed to keep all their goings up and goings down within a vertical range of 50 ft., which is about one one-hundredth of the thickness of our patch of tissue paper, one one-hundred-thousandth of an inch (0.00001). Denizens of the skyscraper city have a vaster range of altitude. You may have an office and do your writing 150 ft. above the sidewalk, and you may even lunch in a restaurant 300 ft. above the curb, and then in your daily life you may boast a vertical range of at least 0.00006 in. on our 8-in. globe.

In this connection we may get the atmosphere, "free air," to do some actual micrometric work for us. The writer has a pocket aneroid barometer, which, as everybody knows, is simply an air pressure gage, that is another of his personal playthings, sometimes used for observations in the elevators of tall buildings. This handy instrument will show a measurable difference in the pressure of the air with a change of elevation of 25 ft., corresponding to 0.005 of the thickness of our sheet of tissue paper, or to 0.000005 in. on the diameter of our 8-in. globe. There are fine instruments used by engineers which will indicate a difference of altitude of 5 ft., which would be 0.001 of the thickness of our bit of tissue paper, or 0.000000125 of the diameter of the earth. The difference of air pressure for a difference of altitude of 5 ft. would be about 0.0025 lb. per square inch.

We are far enough from having discovered and appropriated all the possibilities of air employment for the service of man, and where we do employ it our methods are crude and inefficient as compared with those observed in the large atmospheric phenomena. In the easy way in which nature works the air is made to do its gigantic tasks by means of comparatively slight changes of weight or pressure, of temperature or of humidity. All the changes of the weather may be called compressed air effects, and yet the normal range of atmospheric air pressures as registered by the barometer is not more than half a pound to the square inch, with 1 lb. as the limit of variation, while if man undertakes any compressed-air work such pressures are worthless. A hundred pounds is the common working pressure he employs, while for special purposes it must be thousands of pounds.

Nevertheless it is for us to go on doing the best we can with the air we use, hoping to do better and better, and with little thought as to the disturbances we may cause. We are not likely to joggle the universe much.

Collective humanity, like the individual man, creates for itself—makes out of nothing—many unwarranted worries. Our supplies of coal and iron are going to give out, and what will we do then? Our earth-drawn nitrates for fertilizers are apparently much nearer exhaustion than coal and iron, and now that we turn directly to the atmosphere to abstract a supply a new worry is invented. What will we do when the air fails us?

It is proper to remember that we are not as big in comparison with the earth and its appurtenances as we are apt to think we are, and we have little power to disturb its established arrangements. We may say that—in very round numbers—the total weight of the atmosphere is 5,000,000,000,000,000 tons. That would be—in the same style of numerical rotundity—3,850,000,000,000,000 tons of nitrogen and 1,150,000,000,000,000 tons of oxygen. Now, suppose that we abstracted a million tons, or a hundred million tons, of either, how much would they be likely to be missed?

As a matter of fact we are mechanically abstracting from tho atmosphere very small quantities, comparatively, of both nitrogen and oxygen, and these abstractings may so balance each other that the relative proportions of the two in the atmosphere as a whole may be little disturbed, but even this need not imply that we are reducing the total mass of the air by the amount of its constituents which we take from it by our modern mechanical manipulations. Whatever we seem to get we cannot "get away with" an atom of it We can at the most only borrow, and our borrowings are "returned to stock" in spite of us.

We have to remember nature's automatic and constant restorative processes. The wonderful increase in our industries in the last hundred years has entailed the burning of coal, and now oil, in vast and increasing quantities, and this results in throwing off into the atmosphere incalculable volumes of carbon dioxide, and this alone might ultimately render life on the earth impossible, only that the process does not end with our doings. As we increase the burden of carbon dioxide in the atmosphere by all this active combustion, in addition to the slow combustion in our lungs and in those of all breathing animals, we stimulate plant life by which the carbon is digested out and the oxygen is returned to the air to act as a carrier or go-between over and over again.

It may be safely assumed that the more carbon dioxide there is in the air, the more rapid and luxuriant will plant growth become, especially the rapid growing annual plants upon which we depend for food, these in the aggregate presenting vastly greater air contact surfaces than do the more impressive tree structures. So far as we can see, the result of artificially increasing the production of carbon dioxide, thus stimulating plant growth, is ultimately beneficial to the human race rather than otherwise.

If the matter is sufficiently looked into it will be found that nature's processes provide also for the restoring of nitrogen to the air, if by any other agencies it may be abstracted, so that the balance of atmospheric constituents is maintained from this side also. It is a curious thing that nitrogen, the larger component of the air has been less minutely investigated, and the ramifications of its functions are less definitely known than those of oxygen.

It is well understood that plant growth persistently abstracts nitrates from the soil, and that in successful agriculture these must be artificially replaced, but it is coming to be recognized that there are certain plants which add to the nitrate constituents of the soil, so that these may be made to assist each other; and the further we go in our investigations the more do we discover these automatic compensations.

So far as we may think we decipher at all the ultimate plan of things we may well believe that the atmosphere is made for free and universal use, that any human being may do with any portion of it whatever he will, that the entire human race may do the same, getting whatever benefit and satisfaction they can out of it without the slightest occasion for self reproach or anxiety as to any damages that may be entailed. Fresh air, pure air, will always be accessible to us all whatever we may do to impair it, and from the most crowded haunts of men we need never go many miles to find it, and it is "free as the air "

CHAPTER II

DEFINITIONS AND GENERAL INFORMATION

As we here aim to provide handy rather than minutely precise information for those generally who may have to do with air for mechanical and practical uses, and as we are not proposing to add to the stock of facts and data of the expert scientist, the common English-American standards of weight and measurement will be employed. To have it at hand for immediate use when required, an international conversion table of sufficient scope is given on page 10.

Referring to this table, the figures in the column following that containing the names of the French units are to be used as multipliers in converting those units into their equivalents in English or American measures. Thus 17 meters multiplied by 39.37 equals 669 in. The figures in the column of reciprocals are similarly to be used as multipliers for converting English measures into their French equivalents. Thus 25 sq. in. multiplied by 6.452 equals 161.3 sq. cm.

All measures of length or distance which occur in the book will be given in feet and inches, and weights in pounds avoirdupois. When air, steam or other pressures are referred to they will be pressures in pounds per square inch as indicated upon a common pressure gage, or pressures above that of the normal atmosphere. Absolute pressure is the gage pressure plus the pressure of the atmosphere at the given time and place, this atmospheric pressure being usually taken as 14.7 lb. at sea level. It would be more convenient and would lead to closer approximations in our guess-work calculations if 14.5 were used instead.

Table II., of various pressure equivalents, requires no explanation. It is used in publications of the General Electric Company.

For temperatures the Fahrenheit scale will be used exclusively. A simple formula for converting Centigrade readings to Fahrenheit is:

$$9/5C+32=F.$$

TABLE I

INTERNATIONAL CONVERSION TABLE OF WEIGHTS AND MEASURES

French units	Multiplier	Reciprocal	English and American units
Millimeters	0.03937	25.4	Inches.
Centimeters	0.3937	2.54	Inches.
Meters	39.37	0.0254	Inches.
Meters	3.28083	0.3048	Feet.
Meters	1.0936	0.9144	Yards.
Kilometers	3280.83	0.0003048	Feet.
Kilometers	1093.61	0.0009144	Yards.
Kilometers	0.62137	1.60935	Miles.
Sq. millimeters	0.00155	645.2	Sq. inches.
Sq. centimeters	0.155	6.452	Sq. inches.
Sq. meters	10.764	0.0929	Sq. feet.
Sq. meters	1.196	0.836	Sq. yards.
Sq. kilometers	247.11	0.004045	Acres.
Sq. kilometers	0.386109	2.59	Sq. miles.
Hectares	2.4711	0.4047	Acres.
Cu. centimeters	0.0610	16.3934	Cu. inches.
Cu. meters	35.315	0.02831	Cu. feet.
Cu. meters	1.308	0.7645	Cu. yards.
Centiliters	0.338	2.9585	Fluid ounces.
Liters	61.023	0.01638	Cu. inches.
Liters	0.03531	28.316	Cu. feet.
Liters	33.84	0.0895	Fluid ounces.
Liters	1.0567	0.94636	Quarts.
Liters	0.2642	3.7854	Gallons.
Hectoliters	3.531	0.28316	Cu. feet.
Hectoliters	0.131	7.6335	Cu. yards.
Hectoliters	26.42	0.037854	Gallons (231 cu. in.)
Hectoliters	2.84	0.3521	Bushels (2150.42 cu. in.).
Grams	15.432	0.0647989	Grains.
Grams water	0.03381	29.57	Fluid ounces.
Grams	0.03527	28.3495	Oz. avoirdupois.
Kilograms	35.3	0.028349	Oz. avoirdupois.
Kilograms	2.2046	0.4536	Pounds.
Kilograms	0.00090719	1102.3	Ton (2000 pounds).
Kilograms per sq. centimeter.	14.2226	0.0703	Pounds per sq. in.
Kilogrammeters	7.233	0.1382	Foot-pounds.
Kilos per cheval	2.235	0.4474	Pounds per horse-power.
Kilowatts	1.34	0.746	Horse-power.
Joules	0.7373	1.3427	Foot-pounds.
Calories	3.968	0.25201	B. t. u.

TABLE II

VARIOUS PRESSURE EQUIVALENTS

Inches water pressure	Inches mercury	Ounce per square inch	Pounds per square inch	Pounds per square foot
1.00	0.0736	0.577	0.036	5.19
13.6	1.00	7.84	0.49	70.6
1.73	0.127	1.00	0.0625	9.0
27.7	2.04	16.0	1.00	144.0
0.192	0.0142	0.111	0.00694	1.00

What will 85° Centigrade be by the Fahrenheit scale?

$$(85 \times 9) \div 5 = 153 \text{ and } 153 + 32 = 185$$

For converting Fahrenheit to Centigrade the formula is:

$$5/9(F-32) = C.$$

What is the Centigrade equivalent of 500° Fahrenheit?

$$500-32=468,\ 468 \times 5 = 2340,\ 2340 \div 9 = 260$$

On page 12 is an excellent and most convenient table (Table III) of Centigrade and Fahrenheit readings arranged by Dr. Leonard Waldo, 49 Wall Street, New York, and here reproduced from *Metallurgical and Chemical Engineering.*

Where tables, rules or formulas are given in which numerical multipliers occur, the reciprocals of the multipliers are frequently given also, in parentheses or otherwise, as when the operations indicated are reversed these reciprocals can be used as multipliers, multiplication being generally an easier process than long division.

Atmospheric air is the vapor of a composite liquid which boils and evaporates completely at an extremely low temperature, the vapor having properties and characteristics quite analogous in many particulars to those of steam. While we are quite familiar with water as a solid, as a liquid, and as a vapor or gas, it is only the vapor of liquid air that we normally know about.

Air is roughly stated to be composed of 23 parts by weight of oxygen and 77 parts of nitrogen. By volume the proportions are 21 parts of oxygen and 79 parts of nitrogen. It thus appears that oxygen is somewhat heavier than air while nitrogen is a little lighter, the specific gravity of the former when separated being 1.106 and that of the latter, 0.974, air being 1, and when

TABLE III

CENTIGRADE AND FAHRENHEIT SCALES OF TEMPERATURE

C°	0	10	20	30	40	50	60	70	80	90
	F	F	F	F	F	F	F	F	F	F
−200	−328	−346	−364	−382	−400	−418	−436	−454	.....	......
−100	−148	−166	−184	−202	−220	−238	−256	−274	−292	−310
−0	+32	+14	−4	−22	−40	−58	−76	−94	−112	−130
0	32	50	68	86	104	122	140	158	176	194
100	212	230	248	266	284	302	320	338	356	374
200	392	410	428	446	464	482	500	518	536	554
300	572	590	608	626	644	662	680	698	716	734
400	752	770	788	806	824	842	860	878	896	914
500	932	950	968	986	1004	1022	1040	1058	1076	1094
600	1112	1130	1148	1166	1184	1202	1220	1238	1256	1274
700	1292	1310	1328	1346	1364	1382	1400	1418	1436	1454
800	1472	1490	1508	1526	1544	1562	1580	1598	1616	1634
900	1652	1670	1688	1706	1724	1742	1760	1778	1796	1814
1000	1832	1850	1868	1886	1904	1922	1940	1958	1976	1994
1100	2012	2030	2048	2066	2084	2102	2120	2138	2156	2174
1200	2192	2210	2228	2246	2264	2282	2300	2318	2336	2354
1300	2372	2390	2408	2426	2444	2462	2480	2498	2516	2534
1400	2552	2570	2588	2606	2624	2642	2660	2678	2696	2714
1500	2732	2750	2768	2786	2804	2822	2840	2858	2876	2894
1600	2912	2930	2948	2966	2984	3002	3020	3038	3056	3074
1700	3092	3110	3128	3146	3164	3182	3200	3218	3236	3254
1800	3272	3290	3308	3326	3344	3362	3380	3398	3416	3434
1900	3452	3470	3488	3506	3524	3542	3560	3578	3596	3614
2000	3632	3650	3668	3686	3704	3722	3740	3758	3776	3794
2100	3812	3830	3848	3866	3884	3902	3920	3938	3956	3974
2200	3992	4010	4028	4046	4064	4082	4100	4118	4136	4154
2300	4172	4190	4208	4226	4244	4262	4280	4298	4316	4334
2400	4352	4370	4388	4406	4424	4442	4460	4478	4496	4514
2500	4532	4550	4568	4586	4604	4622	4640	4658	4676	4694
2600	4712	4730	4748	4766	4784	4802	4820	4838	4856	4874
2700	4892	4910	4928	4946	4964	4982	5000	5018	5036	5054
2800	5072	5090	5108	5126	5144	5162	5180	5198	5216	5234
2900	5252	5270	5288	5306	5324	5342	5360	5378	5396	5414
3000	5432	5450	5468	5486	5504	5522	5540	5558	5576	5594
3100	5612	5630	5648	5666	5684	5702	5720	5738	5756	5774
3200	5792	5810	5828	5846	5864	5882	5900	5918	5936	5954
3300	5972	5990	6008	6026	6044	6062	6080	6098	6116	6134
3400	6152	6170	6188	6206	6224	6242	6260	6278	6296	6314
3500	6332	6350	6368	6386	6404	6422	6440	6458	6476	6494
3600	6512	6530	6548	6566	6584	6602	6620	6638	6656	6674
3700	6692	6710	6728	6746	6764	6782	6800	6818	6836	6854
3800	6872	6890	6908	6926	6944	6962	6998	6998	7016	7034
3900	7052	7070	7088	7106	7124	7142	7160	7178	7196	7214
C°	0	10	20	30	40	50	60	70	80	90

C°	F°
1	1.8
2	3.6
3	5.4
4	7.2
5	9.0
6	10.8
7	12.6
8	14.4
9	16.2
10	18.0

F°	C°
1	0.56
2	1.11
3	1.67
4	2.22
5	2.78
6	3.33
7	3.89
8	4.44
9	5.00
10	5.56
11	6.11
12	6.67
13	7.22
14	7.78
15	8.33
16	8.89
17	9.44
18	10.00

Examples: 1347°C = 2444°F + 12°.6F = 2456°.6F : 3367°F = 1850°C + 2°.78C = 1852°.78C liquid air is evaporated the nitrogen boils away first, which is taken advantage of for the commercial segregation of these gases.

Although the combination of oxygen and nitrogen in the air is a mechanical rather than a chemical one, the two gases separating quite easily under certain conditions, and without the intervention of other elements, and although in the life processes of both animals and plants these individual gases are constantly being abstracted in varying proportions, the composition of the atmosphere as a whole is maintained with wonderful precision all over the world; but it must be remembered that there are processes of restoration as well as of abstraction constantly in operation.

Although oxygen comprises less than one-quarter of the atmosphere it is, or has been, more studied and written about and has been considered of much more use and importance than the larger constituent. It would appear that the functions of nitrogen have not been fully and clearly understood, and that, now that attention is called to its importance in the reciprocal economics of animal and vegetable life, it is coming into its own in the minds of men.

The simple conception of the air as composed of oxygen and nitrogen is inadequate and unsatisfying in many respects, because one cannot avoid the knowledge that the air carries with it or in it other ingredients, many and diverse. These contents of it, however, are not generally to be considered as integral parts of it, as they are constantly changing. The most important detail of the air burden is water, sometimes as vapor closely intermingled with and uniformly distributed through it and invisible, and sometimes in minute particles of the same vapor condensed into actual globules of liquid, and seen as clouds if at a distance or as fog or mist if close at hand.

The term "free" air, as distinguished from air which has been artificially compressed, is only used as a matter of convenience and custom. Free air, or air at atmospheric pressure, as was remarked in the preceding chapter, is really compressed air, or air subjected to pressure, as truly as air at 100 lb. pressure is compressed air, and its volume, pressure and temperature vary in accordance with the same laws. By free air, as the term is commonly used, is meant air at atmospheric pressure, and at ordinary temperature, and it is the air as we obtain it when we begin the operation of mechanical air compression. It is free air, or it should be free air, when first admitted to the compressor cylinder, and it is not free air again until it has been compressed, has done its work, generally in the act of re-expansion, and has been exhausted or discharged into the atmosphere, and has become again a part of the mass which encircles the earth.

When we speak of "free" air entering the compressor cylinder we do not by that term fix it in any of its varying characteristics. We know nothing precisely as to its pressure, its volume as related to its weight, its temperature or its humidity The pressure and volume of a given weight or actual quantity of free air may vary with the altitude or location, or with the barometric reading at the place and time; or, again, the volume may vary with the temperature. The temperature may vary with the changes of the seasons, with the time of day, or with the general surroundings. For present purposes it is herein generally assumed that our free air is at the normal sea-level pressure of 14.7 lb., absolute, and at a temperature of 60°.

The temperatures dealt with will usually be the sensible temperatures, or those indicated by the Fahrenheit thermometer, 32° being the melting-point of ice, or the point where water changes from the solid to the liquid state, and 212°, or 180° above this, being where the change from the liquid to the gaseous state occurs. The boiling-point is in practice a constantly variable one, and depends entirely upon the pressure of the atmosphere upon the water, 212° being the boiling-point only at ordinary atmospheric pressure near the sea level. Water may theoretically be made to boil at any temperature above the freezing-point by sufficiently reducing the atmospheric pressure upon its surface.

Absolute temperature by the Fahrenheit scale is the temperature as indicated by the thermometer plus 461°. Thus at 60° by the thermometer the absolute temperature is $60+461=521$. At zero temperature by the thermometer the absolute temperature is $0+461=461$. If the thermometric temperature is, say, 30° below zero, or -30, the absolute temperature will be $-30+461=431$, and so on.

In all questions relating to the volume, pressure or weight of air, whether "compressed" or not, the absolute temperature is an important factor, as the volume of the air will vary directly as the absolute temperature, and the pressure and the actual weight of the air will have changing relations. If the absolute temperature of any body of air is increased, the volume will be increased in the same proportion, the pressure remaining constant. So if the absolute temperature of the air be reduced the volume will be reduced equally with it if the pressure is unchanged

The relations of volume, pressure and temperature of air are thus summarized:

1. The absolute pressure of air varies inversely as the volume when the temperature is constant.

2. The air volume varies inversely as the absolute pressure when the temperature is constant.

3. The absolute pressure varies directly as the absolute temperature when the volume is constant.

4. The volume varies as the absolute temperature when the pressure is constant.

5. The product of the absolute pressure and the volume is proportional to the absolute temperature.

A cubic foot of dry air at atmospheric pressure and at any absolute temperature, Fahrenheit, will weigh 39.819 lb. divided by the absolute temperature. Thus at 60° a cubic foot of air weighs $39.819 \div (60+461) = 0.0764$ lb. So, inversely, the volume of, say, 1 lb. of air at atmospheric pressure and at any absolute temperature may be ascertained by dividing the temperature by 39.819, or by multiplying by its reciprocal, 0.025114.

Thus at 60°, as before, $521 \div 39.819 = 13.084$ cu. ft. or $521 \times 0.025114 = 13.084$ cu. ft.

Table IV shows the weight and volume of atmospheric air at sea-level at different temperatures, and requires no explanation. It will be noticed that the figures of column 3 are the reciprocals of those in column 2, and *vice versa*.

If the temperature and the pressure of air both vary, the constant 2.7093 (reciprocal 0.3691) multiplied by the absolute pressure in pounds, per square inch and divided by the absolute temperature will give the weight of a cubic foot.

What will be the weight of 1 cu. ft. of air at 60 lb. pressure and 100° temperature?

$$2.7093 \times (60+14.7) \div (100+461) = 0.3607 \text{ lb.}$$

The volume in cubic feet of 1 lb. of air may be ascertained by dividing the absolute temperature by the absolute pressure and either dividing it by the constant 2.7093, or, preferably, multiplying it by the reciprocal 0.3691.

TABLE IV.—WEIGHT AND VOLUME OF AIR AT SEA-LEVEL AND AT DIFFERENT TEMPERATURES

Temperature, degrees Fahr.	Weight of 1 cu. ft. in pounds	Volume of 1 lb. in cubic feet
0	0.0863	11.582
10	0.0845	11.834
20	0.0827	12.085
30	0.0811	12.336
32	0.0807	12.386
40	0.0794	12.587
50	0.0779	12.838
60	0.0764	13.089
70	0.0750	13.340
80	0.0736	13.592
90	0.0722	13.843
100	0.0710	14.094
110	0.0697	14.345
120	0.0685	14.596
130	0.0674	14.847
140	0.0662	15.098
150	0.0651	15.350
160	0.0641	15.601
170	0.0631	15.852
180	0.0621	16.103
190	0.0612	16.354
200	0.0602	16.605
210	0.0593	16.856
212	0.0591	16.907

What will be the volume of 1 lb. of air at 75° temperature and 50-lb. gage pressure?

$$(75+461) \div (50+14.7) \times 0.3691 = 3.0577 \text{ cu. ft.}$$

If the temperature of air is changed from one absolute temperature T to another absolute temperature t, the volume remaining constant, the resulting absolute pressure p may be obtained from the original absolute pressure P by the simple proportion:

$$T:t::P:p, \text{ or, } \frac{P \times t}{T} = p.$$

If the air enclosed in an air receiver is at 50-lb. gage, or 64.7 absolute pressure, and at $60° + 461 = 521$ absolute temperature, what will be its absolute pressure if its temperature is raised to $200+461=661°$ absolute temperature?

$$(64.7 \times 661) \div 521 = 82.08 \text{ lb. abs.}$$

If the temperature of air is changed from one absolute temperature T to another absolute temperature t, as before, the pressure in this case remaining constant, the resulting volume v (say in cubic feet) may be obtained from the original volume V by the simple proportion:

$$T:t::V:v, \text{ or } \frac{V \times t}{T} = v$$

If 100 cu. ft. of air at $60+461=521°$ absolute temperature have its temperature raised to $220+461=681°$, absolute, what will then be its volume?

$$(100 \times 681) \div 521 = 130.71 \text{ cu. ft.}$$

Specific Heat.—The unit of heat generally employed in records and computations is that quantity of heat which will raise the temperature of 1 lb. of water 1° F., this being known as the British thermal unit (B.t.u.). One unit of heat if applied to 1 lb. of anything else will not have precisely the same heating effect which it has when applied to water. More heat is required to raise the temperature of 1 lb. of water 1° than is required for any other substance. The heating effect of a unit of heat applied to different substances is found to vary widely, and the special quantity of heat required to raise the temperature of 1 lb. of any substance 1° is known as its specific heat. The specific heat of water being 1, all other specific heats are necessarily fractions. That of air being 0.2377, or less than one-quarter that of water, the same unit of heat which would raise the temperature of 1 lb. of water 1° would raise the temperature of 1 lb. of air more than 4°.

The addition of heat to air, or to any elastic fluid may have either of two effects. It may increase the volume while the pressure remains constant, or it may increase the pressure while the volume remains constant. The specific heat, however, will be quite different in the two cases. The specific heat of air—0.2377, as given above—is its specific heat at constant pressure, and the heat in this case exhibits its effect by increasing the volume of the air. If the air be confined so that there can be no increase of volume, its specific heat is then only 0.1688, or about one-sixth that of water. If heat be applied to air under constant pressure, raising its temperature from the freezing- to the boiling-point of water—from 32° to 212°—the increase in volume will be from 1 to 1.365; and if heat be applied to air at constant volume, raising its temperature as before from 32° to 212°, the increase in absolute pressure will be from 1 to 1.365, the numerical result being alike in the two cases, but the heat expended will be as 0.2377 : 0.1688, or nearly one-half more in one case than in the other.

When air is compressed, or when its volume is reduced by the application of force, the temperature of the air is raised. This phenomenon occurs entirely regardless of the time occupied in the compression; the heating does not follow the compression but is coincident with it. If during the compression the air neither loses nor gains any heat by conduction or radiation to or from any other body, the heat produced by the act of compression remaining in the air and increasing its temperature, then the air is said to be compressed adiabatically, and such compression is adiabatic compression.

When air under pressure is allowed to expand into a larger volume its pressure and its temperature both fall, and if during the operation the air receives no heat from anything outside itself, it is said to expand adiabatically. Adiabatic compression or expansion of air is compression or expansion without the air losing or gaining any heat. The expression "without loss or gain of heat," it will be understood, does not mean the maintaining of the air at a constant temperature, but precisely the reverse of that.

If during compression the air could be kept at a constant temperature, by the abstraction of the heat as fast as it was developed, the air would then be said to be compressed isothermally. In isothermal compression or expansion the air remains at a constant temperature, and it therefore must and does lose or gain heat from some source outside itself throughout the operation.

The rate of increase in the temperature of air during adiabatic compression is not uniform. The temperature rises faster at the beginning and during the earlier stages of the compression than it does when the higher pressures are reached. Thus in compressing air from a pressure of 1 atmosphere to 2 atmospheres by mechanically reducing its volume, the increase in temperature will be greater than in compressing from 2 to 3 atmospheres, and so on. The rate of increase of temperature also varies with the initial temperature. The higher the initial temperature the greater will be the rate of increase of temperature at any point, and throughout the subsequent compression.

The Barometer.—Nature is a compressed-air worker. The routine of natural activities includes, among many other things, the constant and apparently irregular changing of the air-pressure, or the compression and the re-expansion of the air which encloses us. The crudeness, the wastefulness, and the comparative inefficiency of man's methods are indicated by the range of pressures which he is compelled to employ in his manipulations of the air, as compared with the local ranges of atmospheric pressures.

In our industrial operations we have need of pressure gages with an aggregate reach of at least 3000 lb. to the square inch, while practically the entire range of pressures of the compressed air which nature works with in any special locality on the earth is only about a pound, or, say, at sea-level from 29 to 31 in. of mercury. The usual changes for a week at a time may all be within a quarter of a pound of pressure, or say half an inch of mercury, and yet we find it pays to watch and to keep a record of these minute changes, for they go along with the water-carrying and water-distributing functions of the air; and the effects of these, the rain, the fog and then the sunshine, it is desirable to anticipate and to make the most of for our good.

The atmospheric pressure changes being so small, it is necessary to be able to measure them with great minuteness, and so we have the barometer, which is simply a pressure gage, but with some wonderful refinements which give minute precision of record, as spoken of in the preceding chapter.

The diminution of the pressure of air, or its attenuation as altitude increases, makes it necessary to allow for this in determining the dimensions and capacities of air compressors when definite amounts of work are required, and Table V will be of service for this purpose. It is designed to cover the entire range of atmospheric pressures likely to be encountered within the working altitudes, not only above sea-level but for 1000 ft. below it. The table includes for each altitude the height of the mercury column, the corresponding absolute air-pressure in pounds per square inch, the boiling-point of water, the weight of a cubic foot of free air, the percentage of weight of this compared with sea-level air and the number of cubic feet of free air which at each altitude would weigh 1 lb.

TABLE V.—VOLUME AND WEIGHT OF ATMOSPHERIC AIR AT DIFFERENT ALTITUDES

Barometer	Absolute pressure	Altitude	Boiling-point	Weight 1 cu. ft. 60°	Percentage of sea-level weight	Cubic feet per pound
30.93	15.2	−1000	....	.07900	103.40	12.65
30.73	15.1	−800	....	.07848	102.72	12.74
30.52	15.0	−600	213	.07796	102.03	12.82
30.32	14.9	−400	....	.07744	101.36	12.91
30.12	14.8	−200	....	.07693	100.69	12.99
29.91	14.7	200	212	.07640	100.00	13.09
29.71	14.6	200	....	.07589	99.33	13.17
29.50	14.5	400	....	.07536	98.63	13.26
29.30	14.4	600	211	.07484	97.95	13.36
29.10	14.3	800	....	.07432	97.27	13.45
28.90	14.2	1,000	....	.07380	96.59	13.55
28.69	14.1	1,200	210	.07329	95.93	13.64
28.49	14.0	1,400	....	.07277	95.25	13.74
28.28	13.9	1,600	....	.07225	94.96	13.84
28.08	13.8	1,800	209	.07173	93.89	13.94
27.88	13.7	2,000	....	.07120	93.19	14.04
27.67	13.6	2,100	....	.07068	92.51	14.14
27.47	13.5	2,300	208	.07016	91.83	14.25
27.27	13.4	2,500	....	.06965	91.16	14.35
27.06	13.3	2,700	207	.06913	90.48	14.46
26.86	13.2	2,900	....	.06861	89.81	14.57
26.66	13.1	3,100	....	.06809	89.12	14.68
26.45	13.0	3,300	206	.06757	88.44	14.79
26.25	12.9	3,500	....	.06705	87.76	14.91
26.05	12.8	3,700	....	.06652	87.07	15.03
25.84	12.7	4,000	205	.06600	86.38	15.15
25.64	12.6	4,200	....	.06549	85.72	15.27
25.44	12.5	4,400	204	.06497	85.04	15.39
25.23	12.4	4,600	....	.06445	84.36	15.51
25.03	12.3	4,800	....	.06393	83.67	15.64
24.83	12.2	5,000	203	.06341	83.00	15.77
24.62	12.1	5,200	....	.06289	82.31	15.90
24.42	12.0	5,400	202	.06237	81.63	16.03
24.22	11.9	5,600	....	.06185	80.95	16.16
24.01	11.8	5,800	....	.06133	80.23	16.30

TABLE V.—VOLUME AND WEIGHT OF ATMOSPHERIC AIR AT DIFFERENT ALTITUDES.—(*Continued*)

Barometer	Absolute pressure	Altitude	Boiling-point	Weight 1 cu. ft. 60°	Percentage of sea-level weight	Cubic feet per pound
23.81	11.7	6,100	201	.06081	79.59	16.44
23.60	11.6	6,300	....	.06029	78.91	16.58
23.40	11.5	6,500	200	.05977	78.23	16.73
23.20	11.4	6,800	....	.05925	77.55	16.87
22.99	11.3	7,100	199	.05873	76.87	17.02
22.79	11.2	7,300	....	.05821	76.19	17.18
22.59	11.1	7,600	....	.05769	75.51	17.33
22.38	11.0	7,900	198	.05717	74.83	17.48
22.18	10.9	8,100	....	.05665	74.15	17.65
21.98	10.8	8,400	197	.05613	73.47	17.81
21.77	10.7	8,600	....	.05561	72.79	17.98
21.57	10.6	8,900	196	.05509	72.11	18.15
21.37	10.5	9,100	....	.05457	71.42	18.32
21.16	10.4	9,400	195	.05405	70.74	18.50
20.96	10.3	9,600	....	.05353	70.06	18.68
20.76	10.2	9,900	....	.05301	69.38	18.86
20.55	10.1	10,100	194	.05249	68.70	19.05
20.35	10.0	10,400	....	.05198	68.03	19.24
20.15	9.9	10,700	193	.05146	67.35	19.43
19.94	9.8	11,000	....	.05084	66.67	19.63
19.74	9.7	11,200	192	.05041	65.98	19.83
19.53	9.6	11,500	....	.04990	65.31	20.04
19.33	9.5	11,800	191	.04937	64.49	20.25
19.13	9.4	12,100	....	.04886	63.95	20.46
18.93	9.3	12,400	190	.04834	63.27	20.68
18.72	9.2	12,700	....	.04782	62.59	20.91
18.52	9.1	13,000	189	.04730	61.91	21.14
18.31	9.0	13,400	188.5	.04678	61.23	21.37

The Mercury Gage and Its Successors.—In the beginning of modern steam-engine practice, when working steam pressures were very low, and when there was a condenser as an integral part of each engine, it became necessary for the engineer to know both his steam pressure and the tenuity or otherwise of his vacuum, and so steam gages and vacuum gages came to be used. For the want of better means the mercury column was used almost universally both for the steam pressure and for the vacuum.

3

Within the memory of the present writer steamboats were running upon American waters which had open mercury steam gages. The mercury was contained in an inverted siphon composed of iron pipe. One of the vertical pipe ends was connected with the boiler and in the other or open end, which was longer, there was a loosely sliding wooden rod the end of which floated upon the mercury column, and the pressure was indicated by the height of this rod. A similar siphon was used for the vacuum gage, the long end in this case being connected with the condenser and the other end having the floating stick in it the same as the steam gage.

This twofold use of the mercury column developed an anomaly in practice which has survived to this day. For the steam-pressure gage the reading began at zero for normal atmospheric pressure, and as the pressure increased and the rod floating upon the mercury rose, its increasing height above the zero mark was recorded as the steam pressure. For 10 lb. of steam pressure above the atmosphere there would be about 20 in. of mercury, and so on.

For the vacuum gage, however, starting from zero as before, the scale for the mercury column was made to read increasingly as the pressure decreased. Thus from zero, or an absolute pressure of, say, 15 lb., which was near enough for those days, if the absolute pressure was *reduced* by vacuum say, 5 lb., the reading of the mercury gage would not be 5 lb. minus, but 5 lb. plus. We still use the mercury values for our vacuum gages, although we no longer use the mercury, and we also continue to read the gage record inversely.

The use of mercury for either steam or vacuum gages is now practically abandoned, while the readings on the dials of our vacuum gages are still given in inches of mercury. It would seem to be desirable that the equivalents in pounds per square inch should also be given, or, better still, the pounds only. Although the pounds indicated would be pounds of vacuum, or of diminution of pressure, this would be the most convenient in steam-engine practice, as the amount of reduction of pressure by any partial vacuum would then be directly added to the steam pressure on the other side of the piston to give the total effective pressure.

Compressed Air by the Pound.—In statements and computations having to do with the compression, transmission and use of air the basis or unit of quantity is the volume of free air, usually in cubic feet, or air at atmospheric pressure at the time and place under consideration. There are some inconveniences and uncertainties about this practice, because the value of the unit is constantly varying with both the altitude, or normal local pressure, and with the temperature. The actual working volume can always be compared with what would be the volume at sea-level and at an accepted mean temperature, say 60°, but it is an unwelcome operation.

To know the actual quantity of air handled at any time and place would be to work with more certainty. If the actual

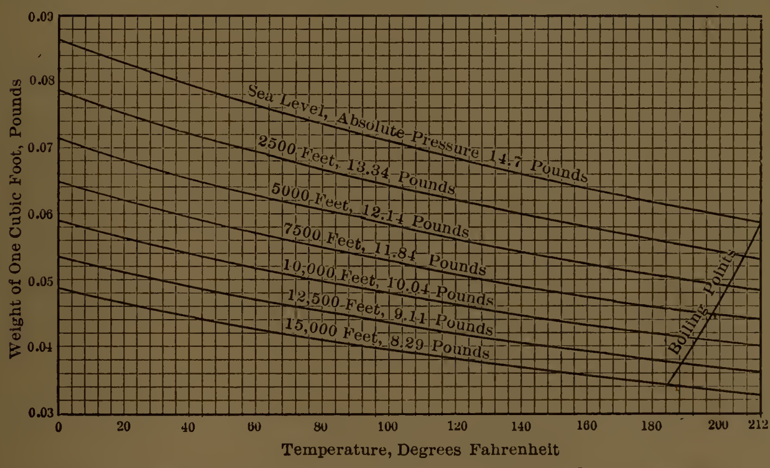


FIG. 2.—Weight of Free Air at Various Altitudes and Temperatures.

weight of the air is known it remains the same whatever the transformations of condition. Pressure, volume, and temperature may all vary, but "a pound's a pound for a' that."

To facilitate the using of the weight basis for records and computations, Table VI and the diagram Fig. 2 have been prepared Computing the items of this table was in detail a simple operation. The weight of a cubic foot of air, whether "compressed" or not, is obtained, as previously explained, by multiplying the constant 2.7093 by the absolute pressure in pounds and dividing the product by the absolute temperature. Thus the weight of 1 cu. ft. of air at an altitude of 10,000 ft. (absolute pressure 10.04 lb) at 80° is:

$$\frac{2.7093 \times 10.04}{80+461} = .0502 \text{ lb.}.$$

The diagram Fig. 2 embodies all the data comprised in the table, while the curve at the right hand, where it cuts the altitude curves, gives the boiling-points of water at these altitudes.

TABLE VI.—WEIGHT OF 1 CU. FT. OF FREE AIR AT VARIOUS ALTITUDES AND TEMPERATURES

Temp. Fahr.	Sea-level abs. press. 14.72	2500 ft. abs. press. 13.34	5000 ft. abs. press. 12.14	7500 ft. abs. press. 11.84	10,000 ft. abs. press. 10.04	12,500 ft. abs. press. 9.11	15,000 ft. abs. press. 8.29
0	0.0863	0.0785	0.0713	0.0648	0.0589	0.0535	0.0487
10	0.0845	0.0768	0.0698	0.0635	0.0577	0.0524	0.0477
20	0.0827	0.0752	0.0683	0.0622	0.0565	0.0513	0.0467
30	0.0811	0.0737	0.0669	0.0609	0.0554	0.0503	0.0457
32	0.0807	0.0734	0.0667	0.0606	0.0551	0.0501	0.0455
40	0.0794	0.0722	0.0656	0.0596	0.0542	0.0493	0.0448
50	0.0779	0.0708	0.0643	0.0585	0.0532	0.0483	0.0439
60	0.0764	0.0695	0.0631	0.0574	0.0522	0.0473	0.0431
70	0.0750	0.0682	0.0619	0.0563	0.0512	0.0465	0.0423
80	0.0736	0.0669	0.0608	0.0552	0.0502	0.0456	0.0415
90	0.0723	0.0657	0.0596	0.0542	0.0493	0.0448	0.0408
100	0.0710	0.0645	0.0586	0.0533	0.0484	0.0440	0.0400
110	0.0697	0.0634	0.0578	0.0523	0.0476	0.0432	0.0393
120	0.0685	0.0623	0.0565	0.0514	0.0466	0.0424	0.0386
130	0.0674	0.0612	0.0556	0.0506	0.0459	0.0417	0.0378
140	0.0662	0.0602	0.0549	0.0497	0.0452	0.0410	0.0373
150	0.0651	0.0592	0.0538	0.0489	0.0444	0.0404	0.0367
160	0.0641	0.0583	0.0529	0.0481	0.0438	0.0397	0.0361
170	0.0631	0.0573	0.0521	0.0473	0.0431	0.0391	0.0358
180	0.0621	0.0565	0.0513	0.0466	0.0424	0.0383	0.0348
190	0.0612	0.0556	0.0504	0.0459	0.0417	0.0379	0.0343
200	0.0602	0.0547	0.0487	0.0452	0.0411	0.0373	0.0338
210	0.0593	0.0539	0.0489	0.0445	0.0405	0.0368	0.0332
212	0.0591	0.0538	0.0488	0.0444	0.0404	0.0366	0.0331

The diagram shows at a glance the differences in weight, or in the actual quantity of air, comprised in a cubic foot of free air at the different elevations. Thus, to take the extremes, the weight of a cubic foot of free air at sea level and at 60° is 0.0764 lb., while at an elevation of 15,000 ft. the weight of the same volume of air at the same temperature is 0.0431, this latter being only 56 per cent. of the former.

FIG. 3.—The Lookout at Lookout Mountain Tunnel.

In compressing this air to, say, 100 lb. local gage pressure, the ratio of compression at sea level would be:

$$\frac{100+14.7}{14.7}=7.8 \text{ atmospheres,}$$

while at 15,000 ft. it would be

$$\frac{100+8.29}{8.29}=13.06 \text{ atmospheres.}$$

This suggests the difference in the work of compression and in the results of compression at different altitudes.

CHAPTER III

THE COMPRESSED-AIR PROBLEM

The general problem of air compression and of the employment of compressed air for the transmission and redevelopment of power wherever it may be required can be stated in simple terms. The crude sketch, Fig. 4, embodies practically all the essentials.

The piston F is fitted to the cylinder E so that we may assume it to move freely and without friction or leakage. It is also supposed to be without weight, and in speaking of it in various positions in the cylinder the lower face of the piston only is referred to.

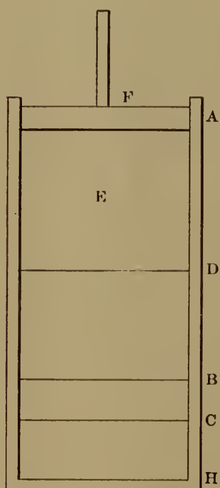


Fig. 4.—The Compressed-air Problem.

The piston being at A, as shown, and the cylinder being full of free air, or air at the normal pressure of 1 atmosphere, and at whatever may be the local temperature at the moment, a sufficient weight is placed upon the piston to force it down into the cylinder and to compress the air contained in it to a pressure of, say, 6 atmospheres.

The theoretical volume being inversely as the absolute pressure, the piston should go down to C, enclosing a space one-sixth of that which the air occupied at the beginning. We find, however, that the piston actually goes down only to B, and the reason is that while the air is being compressed the operation of compression also heats the air, and the rise of temperature is accompanied by an increase of volume, so that the space which it occupies after the compression is considerably larger than it should be upon the assumption that the volume is inversely as the pressure.

Supposing both the piston and the cylinder to be absolute non-conductors of heat, and that the air heated by the compression loses none of this heat of compression, then if the weight which forced the piston down to B, be removed the piston will be driven back to its original position A, and the air contained in the cylinder will have resumed its normal volume, pressure and temperature, and it will have done as much work, or will have exerted as much force, in the expansion as was applied to it in the act of compression.

If, however, while the piston was at B, with the weight upon it sufficient to balance the pressure of 6 atmospheres, the air in some way had given off all its heat of compression and had been cooled to its original temperature, the piston would have descended to C, and the law that the volume varies inversely as the absolute pressure would have held good, for then the initial and the terminal temperatures would have been the same.

We have considered thus far only the compression of the air and what occurs to it in consequence of its change of temperature. We will find that temperature also has its complicating effect when we proceed to use the air for motor purposes, or attempt to obtain through its action a return of the force which has been expended in its compression.

The air being cooled to its original temperature, and the piston being at C, we might expect that upon removing the compressing weight the piston would return to A, its original position. It is found, however, in practice that the piston will return only to D. When it reaches this point the pressure of the air in the cylinder has fallen to its original pressure of 1 atmosphere, and the piston at D is balanced between the equal pressures above and below it.

As the air is heated in the operation of compression so does its temperature correspondingly drop when any expansion or re-expansion occurs, and when the rising piston reaches D the air in the cylinder is already down to atmospheric pressure, because it is then much cooler than when its return movement upward began, and it is solely because of this loss of heat that the piston does not return to A from whence it started.

If while the piston is at D the air can by any means recover all the heat which it is here assumed to have lost while compressed, the piston will return to A as before. This sketch, Fig. 4, is drawn approximately to scale when the initial compression is to 6 atmospheres as here spoken of, and from it we may get a crude idea of the power loss entailed. The distance DA

compared with *CA*, or the distance *DC*, represents the total possible loss of power, with the pressures here assumed, in the compression and re-expansion of the air, theory here taking cognizance of all the conditions.

The use of the weight upon the piston as above for compressing the air, and then the removal of the weight all at once when we wish to allow the air to re-expand is somewhat misleading. In the compressing of the air and in the re-expanding of the air-by its thrust against the piston work is being done, and the weight does not correctly represent the force used. In the beginning of the compression the weight or power required is very small, and

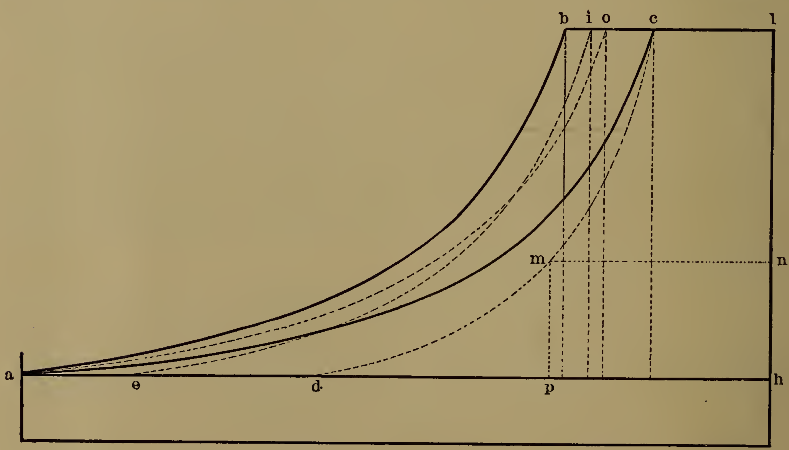


Fig. 5.—Diagram Showing Compression and Re-expansion Effects.

while this power requirement gradually increases it is only at the last moment of the compression that the total weight spoken of is required. It is the same inversely with the re-expansion. When that begins the total weight which opposes the piston is not all at once removed, but is gradually diminished as the piston advances until it all disappears.

We may now refer to the more or less practical indicator diagram Fig. 5, scale 40, which is intended to show more clearly the practical possibilities in the compression and use of air at 75 lb., or 6 atmospheres, absolute. The line *ab* is the adiabatic compression-line, or the line of effective resistance encountered in the compression stroke upon the assumption that no heat is taken away from or is lost by the air during the compression.

The initial temperature of the air being 60°, the final temperature would be about 415°, and the final volume would be 0.28 of the original volume. The line *ac* is the isothermal compression-line, which assumes that all the heat of compression is got rid of just when it is produced, or that the air throughout the compression remains constantly at its initial temperature. The final volume in this cases is 0.1666 of the original volume. Remembering that these lines, *ab* and *ac*, represent the compression of the same initial volume of air, it is evident that there is quite a difference in the amount of power employed in the two cases, and herein lies the loss, or the possibility of loss, of power in the operation of compression. The mean effective pressure or resistance of the air for the stroke upon the adiabatic line *abl* is 35.36 lb., while the mean effective pressure for the isothermal compression-line *acl* is but 27 lb., or only 76 per cent of the former.

The comparison should, however, be reversed. The adiabatic mean effective pressure is 131 per cent. of the isothermal mean effective pressure:

$$27:35.36::1:1.31.$$

and this 31 per cent. is, of course, the additional or, as we might say, the unnecessary power employed, assuming isothermal compression to be attainable.

Neither of these compression-lines, *ab* or *ac*, is possible in practice. Air cannot be compressed without losing some of its heat during compression, so that the actual compression-line must always fall within or below the line *ab*. On the other hand, it is equally impossible to abstract all the heat from the air coincidently with the appearance of that heat, so that the actual compression-line must always fall outside or above the line *ac*. The dotted line *ao* represents the mean of the adiabatic and the isothermal lines, and the best attainable practice may be expected to run outside this line *ao*. The mean effective pressure for the line *aol*, which includes the expulsion or delivery of the air in addition to its compression is about 31 5 lb., or still nearly 17 per cent. in excess of the M.E.P. for the line *acl*.

In all these comparisons for efficiency the actual compression and delivery line is always to be compared with the isothermal line *acl*, because that is the ideal line for compression without sacrifice of power, and because the terminal volume *cl* is the volume actually available for use, no matter how economically or how wastefully the air may have been compressed. Though at the completion of the compression stroke there is always some of the heat of compression remaining in the air, and though its volume at the instant of delivery is always greater than *cl*, that heat may be said to be always lost in the transmission of the air, or in its storage, and the available volume is never practically above *cl*.

After the cooling and contraction of the air compressed comes the question of the cost of transmitting and distributing it to the points where it is to do its work. To cause the air to flow through the pipes there must be some excess of pressure at the first end, a decrease of pressure as the air advances and a greater or less reduction of effective pressure at the delivery end. But the ultimate loss of power in transmission has been greatly exaggerated.

The actual truth is that there is very little loss of power through the transmission of compressed air in suitable pipes to a reasonable distance, and the reasonable distance is not a short one. With pipes of proper size, and in good condition, air may be transmitted, say, ten miles, with a loss of pressure of less than 1 lb. per mile. If the air were at 80 lb. gage, or 95 lb. absolute, upon entering the pipe, and 70 lb. gage, or 85 lb. absolute, at the other end, there would be a loss of a little more than 10 per cent. in absolute pressure, but at the same time there would be an increase of volume of 11 per cent. to compensate for the loss of pressure, and the loss of available power would be less than 3 per cent. With higher pressures still more favorable results could be shown.

Having compressed the air and conveyed it to the point where we wish to use it, we may turn again to Fig. 5, and see what we will be able to do with the air. The air may be used in various ways with widely different economic results, and little ingenuity would be required to develop considerable losses, if losses were what we were after. Having the volume *cl*, and using it in a cylinder of suitable capacity, with a point of cut-off which would allow the air to expand down to about 1 atmosphere before release, the adiabatic expansion line, or the lowest line that the air could make would be the line *cd*, and the total loss in the use of the air, as compared with the power cost of compressing it, would be the difference between the areas *aolh* and *lcdh*, the latter being 66 per cent. of the former.

The temperature of the air at *c*, where the expansion begins, being assumed to be 60°, the cooling of the air which accompanies its expansion will bring the temperature far down the scale when *d* is reached, *d* being the end of the cylinder wherein the expansion takes place. The theoretical temperature of the air at the end of the stroke would be about −150°. The actual temperature in these cases is never found as low as the theoretical temperature, because the air receives heat from the cylinder and from the walls of the passages with which it comes in contact; but it is usually still cold enough to cause serious inconvenience in practice, and this cooling of the air may in many cases be prohibitive, entirely regardless of the economy of the case. The air always contains moisture, the amount varying with the surrounding meteorological conditions, and as the air becomes attenuated and so intensely cold the water is rapidly frozen in the passages, and soon chokes them up and stops the operation of the motor. The prevention or the circumvention of the freezing up of air apparatus is an additional complication of the compressed-air problem to be considered later.

The trouble from the freezing up naturally suggests the heating of the air before it is used. The heating or re-heating of the air, where it is practised, not only brings us out of our trouble about the freezing up, but it increases the volume of the air and its consequent available power at a very slight expense for the heating. If the volume of air *cl*, being now at 60°, be passed through a suitable heater and its temperature raised to 300°, its volume will then be *il*, instead of *cl*, or 0.2434 instead of 0.1666, an increase of volume of about 50 per cent. In practice, to insure a temperature of 300° in the cylinder at the beginning of the expansion, it will be necessary to heat the air considerably above that temperature, say to 400°, as the air loses its heat very rapidly. If now we use this re-heated air, the volume *cl*, becoming *il*, and expanding this air down to *e*, supposing the temperature at *i* to be 300° the final theoratical temperature will be about zero. The actual temperature, it is pretty certain, will not be below the freezing-point, and all our trouble about the freezing up of the passages will have disappeared, and the power realized per volume of free air used will have been much increased. It seems to be quite practicable in many cases, by effective cooling of the air during its compression, and by re-heating it before its re-expansion, to bring the expansion-line *ic* to enclose with the admission-line an area not less than that enclosed by the compression-line *as*, and then the losses will be those attributable to the clearances and to friction, while there will be the slight additional charge for the re-heating.

CHAPTER IV

TABLES AND DIAGRAMS FOR COMPUTATIONS IN AIR-COMPRESSION

The tables and diagrams presented in this chapter should be of service to those having to do with air-compression, or with compressed air used for motor purposes, who wish to reach approximately reliable results by simple and familiar methods.

The actual compression of air for industrial purposes and by the methods still most generally employed, that is in cylinders with reciprocating pistons, is practically adiabatic in each specific case, and in investigating the power requirements, the actual power consumption and other conditions or results of such compression, it is necessary to know as a basis the pressures, volumes and temperatures involved, and also the mean effective pressures or resistances occurring, and the facilities here presented should make the work easy. The tables are applicable to all cases of single-stage compression, and to all the separate stages of compound- or multi-stage compression.

When other and entirely different, and perhaps equally efficient, devices are employed for compressing air, such as turbo-compressors, hydraulic compressors, etc., it is still convenient to refer to the work of the reciprocating compressor for the comparision and valuation of results accomplished.

Pressures, Volumes and Temperatures.—Referring to Table VII and its corresponding diagram, Fig. 6, it will be seen that if we first ascertain the ratio of the initial to the terminal (or *vice versa*) of either the pressures, the volumes or the temperatures of air compressed and delivered without loss or gain of heat, adiabatic compression, the ratio of the other two particulars are easily determined and then having the ratio in either case, with the actual pressure, volume or temperature at one end, either the beginning or the completion, of the act of compression (or of re-expansion if the air is being used for motor purposes), the corresponding pressures, volumes or temperatures for the other end of the operation also will be known.

TABLE VII.—RATIOS AND RECIPROCALS OF PRESSURE, VOLUME AND TEMPERATURE IN THE ADIABATIC COMPRESSION OF AIR

Ratio	Reciprocal	A	Reciprocal	B	Reciprocal	C	Reciprocal	D	Reciprocal	E	Reciprocal	F	Reciprocal.
0.01	100.0	0.0380	26.315	0.0131	76.259	0.0015	666.66	0.0075	133.33	0.1528	6.554	0.2633	3.799
0.02	50.0	0.0621	16.103	0.0252	39.682	0.0041	243.90	0.0157	63.69	0.2027	4.933	0.3219	3.106
0.03	33.33	0.0829	12.062	0.0369	27.102	0.0072	138.88	0.0241	41.49	0.2392	4.180	0.3620	2.762
0.04	25.0	0.1017	9.832	0.0483	20.704	0.0108	92.592	0.0327	30.58	0.2689	3.718	0.3935	2.541
0.05	20.0	0.1191	8.396	0.0596	16.778	0.0147	68.027	0.0414	24.15	0.2946	3.394	0.4198	2.382
0.06	16.66	0.1356	7.374	0.0708	14.039	0.0190	52.631	0.0503	19.88	0.3173	3.151	0.4425	2.259
0.07	14.28	0.1513	6.609	0.0819	12.210	0.0237	42.194	0.0593	16.86	0.3379	2.959	0.4627	2.161
0.08	12.5	0.1663	6.013	0.0928	10.775	0.0286	34.964	0.0683	14.64	0.3568	2.803	0.4810	2.079
0.09	11.11	0.1808	5.531	0.1037	9.643	0.0337	29.675	0.0774	12.92	0.3744	2.671	0.4977	2.009
0.10	10.0	0.1949	5.131	0.1145	8.733	0.0391	25.575	0.0866	11.54	0.3908	2.558	0.5131	1.948
0.11	9.091	0.2085	4.796	0.1253	7.981	0.0447	22.371	0.0958	10.43	0.4063	2.461	0.5275	1.895
0.12	8.333	0.2218	4.508	0.1359	7.358	0.0505	19.802	0.1051	9.522	0.4210	2.375	0.5410	1.848
0.13	7.692	0.2348	4.259	0.1466	6.821	0.0566	17.668	0.1144	8.741	0.4350	2.298	0.5537	1.806
0.14	7.143	0.2475	4.040	0.1572	6.361	0.0628	15.923	0.1238	8.077	0.4484	2.230	0.5657	1.767
0.15	6.667	0.2599	3.847	0.1677	5.963	0.0692	14.451	0.1332	7.507	0.4612	2.168	0.5771	1.732
0.16	6.25	0.2721	3.675	0.1782	5.611	0.0758	13.192	0.1427	7.007	0.4735	2.112	0.5880	1.700
0.17	5.882	0.2841	3.519	0.1887	5.299	0.0825	12.121	0.1522	6.570	0.4853	2.066	0.5984	1.671
0.18	5.556	0.2959	3.380	0.1991	5.022	0.0894	11.185	0.1617	6.188	0.4968	2.012	0.6084	1.643
0.19	5.263	0.3074	3.253	0.2095	4.773	0.0965	10.362	0.1713	5.837	0.5079	1.968	0.6180	1.618
0.20	5.0	0.3188	3.136	0.2199	4.547	0.1037	9.643	0.1809	5.527	0.5186	1.930	0.6273	1.594
0.21	4.762	0.3301	3.029	0.2302	4.344	0.1111	9.001	0.1905	5.249	0.5290	1.890	0.6362	1.571
0.22	4.545	0.3412	2.930	0.2405	4.157	0.1186	8.431	0.2001	4.997	0.5392	1.854	0.6448	1.551
0.23	4.348	0.3521	2.840	0.2508	3.987	0.1263	7.917	0.2098	4.766	0.5490	1.821	0.6532	1.530
0.24	4.167	0.3629	2.755	0.2610	3.831	0.1341	7.457	0.2195	4.555	0.5586	1.792	0.6613	1.512
0.25	4.0	0.3736	2.676	0.2712	3.687	0.1420	7.042	0.2293	4.361	0.5680	1.760	0.6692	1.494
0.26	3.846	0.3841	2.603	0.2814	3.553	0.1501	6.662	0.2390	4.182	0.5772	1.732	0.6768	1.477
0.27	3.704	0.3946	2.534	0.2916	3.429	0.1583	6.317	0.2488	4.019	0.5861	1.706	0.6843	1.461
0.28	3.571	0.4049	2.469	0.3018	3.313	0.1666	6.000	0.2586	3.867	0.5949	1.681	0.6915	1.446
0.29	3.448	0.4151	2.409	0.3119	3.206	0.1750	5.701	0.2684	3.725	0.6035	1.657	0.6986	1.431
0.30	3.333	0.4252	2.351	0.3220	3.105	0.1836	5.446	0.2783	3.593	0.6119	1.634	0.7055	1.417
0.31	3.226	0.4353	2.297	0.3321	3.011	0.1922	5.203	0.2881	3.471	0.6201	1.611	0.7122	1.404
0.32	3.125	0.4452	2.246	0.3422	2.922	0.2010	4.975	0.2980	3.355	0.6282	1.591	0.7188	1.391
0.33	3.03	0.4550	2.197	0.3522	2.839	0.2099	4.764	0.3079	3.247	0.6361	1.572	0.7252	1.378
0.34	2.941	0.4648	2.151	0.5623	2.760	0.2189	4.568	0.3178	3.146	0.6439	1.553	0.7315	1.365
0.35	2.857	0.4744	2.108	0.3723	2.686	0.2281	4.384	0.3278	3.050	0.6516	1.534	0.7377	1.355
0.36	2.778	0.4840	2.066	0.3823	2.615	0.2373	4.214	0.3377	2.961	0.6591	1.517	0.7438	1.344
0.37	2.703	0.4935	2.026	0.3923	2.549	0.2466	4.055	0.3477	2.876	0.6665	1.500	0.7497	1.333
0.38	2.632	0.5030	1.988	0.4023	2.485	0.2561	3.904	0.3577	2.795	0.6738	1.484	0.7555	1.323
0.39	2.564	0.5124	1.949	0.4122	2.426	0.2656	3.765	0.3677	2.719	0.6810	1.468	0.7612	1.313
0.40	2.5	0.5216	1.917	0.4222	2.368	0.2752	3.633	0.3777	2.647	0.6881	1.453	0.7668	1.304
0.41	2.439	0.5309	1.883	0.4321	2.314	0.2850	3.509	0.3878	2.578	0.6951	1.438	0.7723	1.294
0.42	2.381	0.5400	1.851	0.4420	2.262	0.2948	3.392	0.3978	2.513	0.7019	1.424	0.7777	1.285
0.43	2.326	0.5491	1.819	0.4519	2.212	0.3047	3.281	0.4079	2.451	0.7087	1.411	0.7831	1.276
0.44	2.273	0.5582	1.791	0.4618	2.165	0.3148	3.176	0.4180	2.392	0.7154	1.397	0.7883	1.268
0.45	2.222	0.5672	1.763	0.4716	2.122	0.3249	3.084	0.4281	2.336	0.7220	1.385	0.7934	1.260
0.46	2.174	0.5761	1.735	0.4815	2.076	0.3351	2.984	0.4382	2.282	0.7285	1.372	0.7985	1.252
0.47	2.128	0.5850	1.709	0.4913	2.035	0.3454	2.895	0.4483	2.230	0.7349	1.360	0.8035	1.244
0.48	2.083	0.5938	1.684	0.5012	1.979	0.3558	2.810	0.4585	2.181	0.7412	1.349	0.8084	1.237
0.49	2.041	0.6025	1.659	0.5110	1.957	0.3663	2.730	0.4686	2.134	0.7475	1.337	0.8133	1.229
0.50	2.0	0.6112	1.636	0.5208	1.920	0.3768	2.654	0.4788	2.091	0.7537	1.326	0.8180	1.222

TABLE VII.—RATIOS AND RECIPROCALS OF PRESSURE, VOLUME AND TEMPERATURE IN THE ADIABATIC COMPRESSION OF AIR.—(*Continued*)

Ratio	Reciprocal	A	Reciprocal	B	Reciprocal	C	Reciprocal	D	Reciprocal	E	Reciprocal	F	Reciprocal
0.51	1.961	0.6199	1.613	0.5306	1.884	0.3875	2.581	0.4890	2.045	0.7598	1.316	0.8227	1.215
0.52	1.923	0.6285	1.591	0.5404	1.850	0.3982	2.511	0.4992	2.003	0.7658	1.305	0.8274	1.208
0.53	1.887	0.6371	1.568	0.5502	1.817	0.4091	2.443	0.5094	1.963	0.7718	1.295	0.8320	1.201
0.54	1.852	0.6456	1.549	0.5599	1.786	0.4200	2.381	0.5196	1.924	0.7777	1.285	0.8365	1.195
0.55	1.818	0.6540	1.529	0.5697	1.755	0.4310	2.320	0.5298	1.887	0.7836	1.276	0.8409	1.189
0.56	1.786	0.6625	1.509	0.5794	1.725	0.4420	2.262	0.5401	1.851	0.7893	1.266	0.8453	1.183
0.57	1.754	0.6708	1.491	0.5892	1.697	0.4532	2.206	0.5503	1.817	0.7951	1.257	0.8497	1.177
0.58	1.724	0.6792	1.472	0.5989	1.669	0.4644	2.153	0.5606	1.783	0.8007	1.248	0.8540	1.171
0.59	1.695	0.6875	1.454	0.6086	1.643	0.4757	2.102	0.5709	1.751	0.8063	1.240	0.8582	1.165
0.60	1.667	0.6957	1.437	0.6183	1.613	0.4871	2.051	0.5812	1.720	0.8119	1.231	0.8624	1.159
0.61	1.639	0.7039	1.420	0.6280	1.592	0.4986	2.005	0.5914	1.691	0.8174	1.223	0.8666	1.153
0.62	1.613	0.7121	1.404	0.6377	1.568	0.5101	1.960	0.6018	1.661	0.8228	1.215	0.8706	1.148
0.63	1.587	0.7203	1.388	0.6474	1.544	0.5218	1.910	0.6121	1.633	0.8228	1.207	0.8747	1.143
0.64	1.563	0.7284	1.372	0.6570	1.522	0.5335	1.874	0.6224	1.606	0.8335	1.199	0.8787	1.138
0.65	1.538	0.7364	1.358	0.6667	1.500	0.5452	1.835	0.6327	1.580	0.8388	1.192	0.8827	1.132
0.66	1.515	0.7445	1.343	0.6763	1.478	0.5571	1.795	0.6431	1.554	0.8441	1.184	0.8866	1.128
0.67	1.493	0.7524	1.328	0.6860	1.457	0.5690	1.757	0.6534	1.530	0.8493	1.177	0.8904	1.123
0.68	1.471	0.7604	1.315	1.6956	1.437	0.5810	1.721	0.6638	1.506	0.8544	1.170	0.8943	1.118
0.69	1.449	0.7683	1.301	0.7052	1.418	0.5931	1.686	0.6742	1.483	0.8595	1.163	0.8981	1.113
0.70	1.429	0.7762	1.288	0.7148	1.399	0.6052	1.652	0.6846	1.461	0.8646	1.156	0.9018	1.108
0.71	1.408	0.7841	1.275	0.7245	1.380	0.6174	1.619	0.6950	1.438	0.8696	1.149	0.9055	1.104
0.72	1.389	0.7919	1.263	0.7341	1.359	0.6297	1.588	0.7054	1.417	0.8746	1.143	0.9092	1.099
0.73	1.370	0.7997	1.250	0.7436	1.344	0.6420	1.557	0.7158	1.397	0.8795	1.137	0.9128	1.095
0.74	1.351	0.8075	1.238	0.7532	1.327	0.6545	1.527	0.7262	1.375	0.8844	1.130	0.9165	1.091
0.75	1.333	0.8152	1.227	0.7628	1.311	0.6669	1.499	0.7366	1.357	0.8893	1.124	0.9200	1.087
0.76	1.316	0.8229	1.215	0.7724	1.294	0.6795	1.471	0.7471	1.338	0.8941	1.118	0.9236	1.082
0.77	1.299	0.8306	1.204	0.7819	1.278	0.6921	1.444	0.7575	1.321	0.8989	1.112	0.9271	1.078
0.78	1.282	0.8382	1.193	0.7915	1.263	0.7048	1.418	0.7680	1.302	0.9036	1.106	0.9305	1.074
0.79	1.266	0.8459	1.182	0.8010	1.248	0.7176	1.393	0.7785	1.284	0.9083	1.100	0.9340	1.070
0.80	1.25	0.8534	1.171	0.8106	1.233	0.7304	1.369	0.7889	1.267	0.9130	1.095	0.9374	1.066
0.81	1.235	0.8610	1.161	0.8201	1.219	0.7433	1.345	0.7994	1.251	0.9176	0.089	0.9408	1.062
0.82	1.22	0.8685	1.151	0.8296	1.205	0.7562	1.322	0.8099	1.234	0.9222	1.084	0.9441	1.059
0.83	1.205	0.8761	1.140	0.8392	1.191	0.7692	1.300	0.8204	1.219	0.9268	1.079	0.9474	1.055
0.84	1.19	0.8835	1.131	0.8487	1.178	0.7823	1.278	0.8309	1.203	0.9313	1.073	0.9507	1.051
0.85	1.176	0.8910	1.122	0.8582	1.165	0.7955	1.257	0.8414	1.188	0.9358	1.068	0.9540	1.048
0.86	1.163	0.8984	1.113	0.8678	1.152	0.8087	1.236	0.8519	1.173	0.9403	1.063	0.9572	1.044
0.87	1.149	0.9058	1.104	0.8772	1.139	0.8220	1.216	0.8625	1.159	0.9448	1.058	0.9605	1.041
0.88	1.136	0.9132	1.095	0.8866	1.127	0.8353	1.197	0.8730	1.145	0.9492	1.053	0.9636	1.037
0.89	1.124	0.9206	1.086	0.8961	1.115	0.8487	1.178	0.8835	1.131	0.9536	1.048	0.9668	1.034
0.90	1.111	0.9279	1.077	0.9056	1.104	0.8621	1.159	0.8941	1.118	0.9579	1.043	0.9699	1.031
0.91	1.099	0.9352	1.069	0.9151	1.093	0.8757	1.142	0.9047	1.105	0.9623	1.039	0.9730	1.027
0.92	1.087	0.9425	1.061	0.9245	1.081	0.8892	1.124	0.9152	1.092	0.9666	1.034	0.9761	1.024
0.93	1.075	0.9498	1.052	0.9340	1.070	0.9029	1.107	0.9258	1.080	0.9708	1.030	0.9792	1.021
0.94	1.064	0.9570	1.045	0.9434	1.059	0.9166	1.091	0.9364	1.068	0.9751	1.025	0.9822	1.019
0.95	1.053	0.9642	1.037	0.9529	1.049	0.9303	1.074	0.9470	1.056	0.9793	1.021	0.9853	1.014
0.96	1.042	0.9714	1.029	0.9623	1.039	0.9441	1.059	0.9576	1.044	0.9835	1.016	0.9882	1.012
0.97	1.031	0.9786	1.021	0.9717	1.029	0.9580	1.043	0.9682	1.032	0.9877	1.012	0.9812	1.009
0.98	1.020	0.9858	1.014	0.9812	1.019	0.9720	1.029	0.9788	1.021	0.9918	1.008	0.9942	1.006
0.99	1.01	0.9929	1.007	0.9906	1.009	0.9860	1.014	0.9894	1.011	0.9959	1.004	0.9971	1.003
1.00	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1 0	1.0	1.0	1.0	1.0

Our first and most easily accessible knowledge concerning any specific operation of compression is as to the pressures at the beginning and at the end of the compression. Thus, say that we are working at or near sea-level, or taking in the air at an absolute pressure of 1.47 lb. and delivering the air at 80 lb. gage, the absolute pressure then being $80+14.7=94.7$ lb.; the ratio of initial to terminal pressure will be $14.7 \div 94.7=0.1552$. With these pressures at the opposite ends of the stroke the ratio will, of course, be the same whether compressing the air in the ordinary reciprocating compressor or expanding the air in doing work in an air engine or motor, only in one case the ratio as given would be the convenient multiplier in our computations and in the other case the reciprocal would apply, so that the reciprocal for every ratio is provided in the table as here given, for practical working purposes.

The table is not entirely new, the principal elements of it having been computed more than 30 years ago by the late Richard H. Buel. The reciprocals of all the ratios have been added and the chart has been entirely plotted from the table, the smoothness of the curves in the plotting attesting the accuracy, or at least the consistency, of the figures.

The several curves of the chart are designated by the same letters as the columns of the table to which they correspond, adiabatic compression, or expansion, being assumed throughout. The several series of ratios, whether in the curves of the chart, or in the columns of the table are as follows:

A = Ratios of initial and terminal volumes of air for given ratios of absolute pressures.

B = Ratios of initial and terminal volumes of saturated steam for given ratios of absolute pressures.

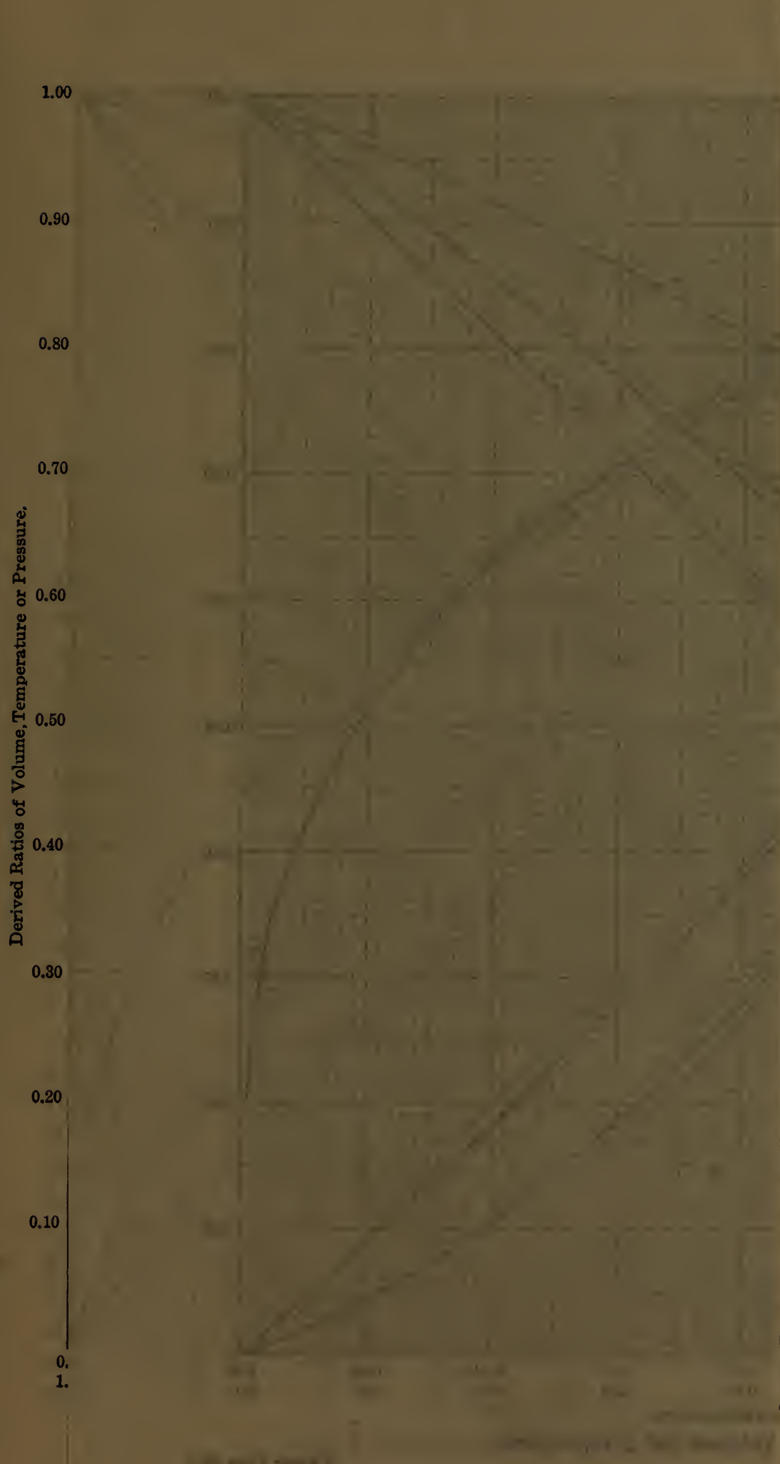
C = Ratios of initial and terminal absolute pressures of air for given ratios of volumes.

D = Ratios of initial and terminal absolute pressures of saturated steam for given ratios of volumes.

E = Ratios of initial and terminal absolute temperatures of air for given ratios of volumes.

F = Ratios of initial and terminal absolute temperatures of air for given ratios of absolute pressures.

The formulas follow by which the ratios in the several columns were computed, $\frac{1}{R}$ being the ratio in the first column of the table.



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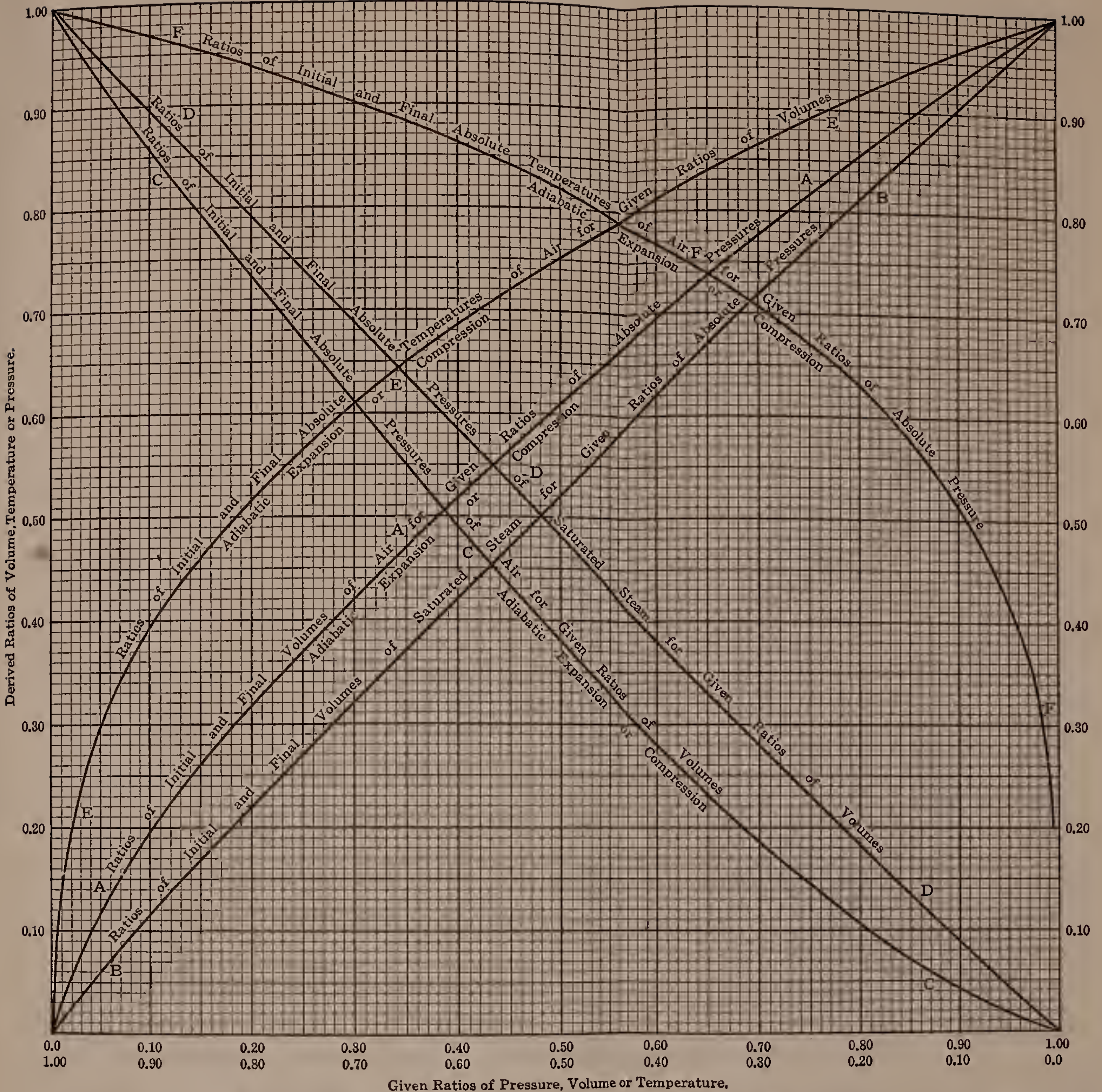

Fig. 6.—Ratios of Initial and Final Pressures, Volumes and Temperatures.

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$A \ldots\ldots\ldots\ldots \left(\frac{1}{R}\right)^{0.71}$

$B \ldots\ldots\ldots\ldots \left(\frac{1}{R}\right)^{\frac{16}{17}}$

$C \ldots\ldots\ldots\ldots \left(\frac{1}{R}\right)^{1.408}$

$D \ldots\ldots\ldots\ldots \left(\frac{1}{R}\right)^{\frac{17}{16}}$

$E \ldots\ldots\ldots\ldots \left(\frac{1}{R}\right)^{0.408}$

$F \ldots\ldots\ldots\ldots \left(\frac{1}{R}\right)^{0.29}$

To anyone appreciating and capable of intelligently using the table, any explanation of methods of procedure would be superfluous. The table, although somewhat voluminous is essentially simple, as is also the chart which embodies it in every particular.

In the chart the given or known ratios, from which the desired results are to be obtained, are noted horizontally on the base-line, and the ratio desired is then indicated by the vertical distance from the base-line at the point noted to the curve of ratios required in the particular case, the scale for the readings being at the side of the chart The curves *A*, *B* and *E* start from the lower left-hand corner, and the curves *C*, *D* and *F* from the right-hand corner, the base-line being double figured for facility of reading in either direction.

As illustrating the use of the chart, say that we have an air-compressor working at an altitude of 2000 ft. above sea-level compressing 900 cu. ft. of air per minute and delivering it into the receiver at 75-lb. gage, pressure. The normal atmospheric pressure at 2000 ft. being, say, 13.7 lb., and the absolute delivery pressure then being $75+13.7=88.7$ lb., the pressure ratio will be $13.7 \div 88.7=0.1544$ To ascertain now the delivery volume of the air compressed, we take this ratio, the first two figures of it as they stand and the other two figures by approximating between the lines, and locate it on the base-line reading from the left-hand corner. Taking 15 spaces and a little less than half of the next space we note the vertical distance from this point up to curve *A*, and this we find to be 0.26 with a little more than half of the next vertical space, say, 0.26 5/8, or, decimally, 0.2662, and this being the ratio of the terminal to the initial volume, the actual terminal volume per minute will be $900 \times 0.2662 = 239.58$ cu. ft. at 75 lb. gage.

The volume thus ascertained is, of course, the volume actually delivered and at the moment of delivery, assuming that the compression was perfectly adiabatic throughout, or that the body of air, while being compressed, neither lost nor gained any heat by conduction, radiation or otherwise; and, as a matter of fact, this is nearly the actual state of affairs in ordinary compression. If, during compression, the air could be maintanied at constant temperature the ratios of volumes would then be practically the same as the ratios of pressures, and the "curve" for this would be a straight diagonal from the lower left-hand to the upper right-hand corner.

To ascertain now the ratio of temperatures as the result of our compression the operation is similar, except that in this case we use curve F instead of curve A. Taking our ratio of pressure 0.1544, with which we started, but this time reading from the right-hand corner we find the vertical reading to curve F to be 0.58, or so little above that the excess is negligible. We nete in the table that the riciprocal of 0.58 is 1.724, and this it will be convenient to use here. If the initial temperature of the air by the thermometer was 60° then the absolute initial temperature would be $60+461=521$, and the terminal temperature would be $521 \times 1.724 = 898$ absolute, or $898-461=473°$ by the thermometer.

If we take our ratio of volumes 0.2662, as previously ascertained and, reading from the left-hand corner, note the vertical reading to curve E, the ratio of temperatures will be found to be the same as that obtained above from curve F.

If we take any horizontal line of the chart, or any horizontal line arbitrarily placed upon the chart, and note the points where curves E and F cut it then the distance from curve E to the left-hand vertical boundary will be the ratio of volumes for that temperature ratio, and the distance from curve F to the right-hand boundary will be the ratio of pressures, so that all particulars embodied in curves A and C are really obtainable from curves E and F.

Mean Effective Pressures in Compression or Expansion of Air.—The following table, VIII, and diagram, Fig. 7, in connection with the preceding, offer special facilities for power computations in connection with the compression or the working expansion of air. The former table and diagram dealt with the interdependent ratios of pressure, volume and temperature, either of which being given the others were readily derivable therefrom.

In the table and diagram here presented the given ratio is that of the volumes at the beginning and the end of the stroke. From that is readily determined the mean effective pressure ratio, either during the entire stroke or during compression or expansion only. On the diagram, the given ratio of volumes is read upon the base-line and the required ratio of the mean effective pressures is then indicated by the vertical height from the base-line at the point thus indicated up to the curve representing the required data. It is assumed that no further explanation is required.

In computations relating to the power developed in the working expansion of air in a reciprocating engine or motor, the ratio of volumes is the particular most readily available, as that is determined by the point of cut-off, but in air compression we first know the ratio of pressures, and the ratio of volumes is derivable from this by the assistance of the previous table or diagram, so that both tables or diagrams are necessary to constitute a complete equipment, and they should be kept together. Of course only absolute pressures are dealt with, and when mean effective pressures are obtained, these also are absolute, and the actual working mean effective is obtained by deducting the pressure of the atmosphere. In compound compression or expansion, for the high-pressure cylinder there must be subtracted the total absolute back pressure, which of course must be much greater than that of the atmosphere alone.

The several columns of the table designated by capital letters, and the corresponding lines of the diagram, represent the following ratios of mean to initial, or terminal, total pressures, the given ratio in each case being the ratio of volumes.

FOR ENTIRE STROKE

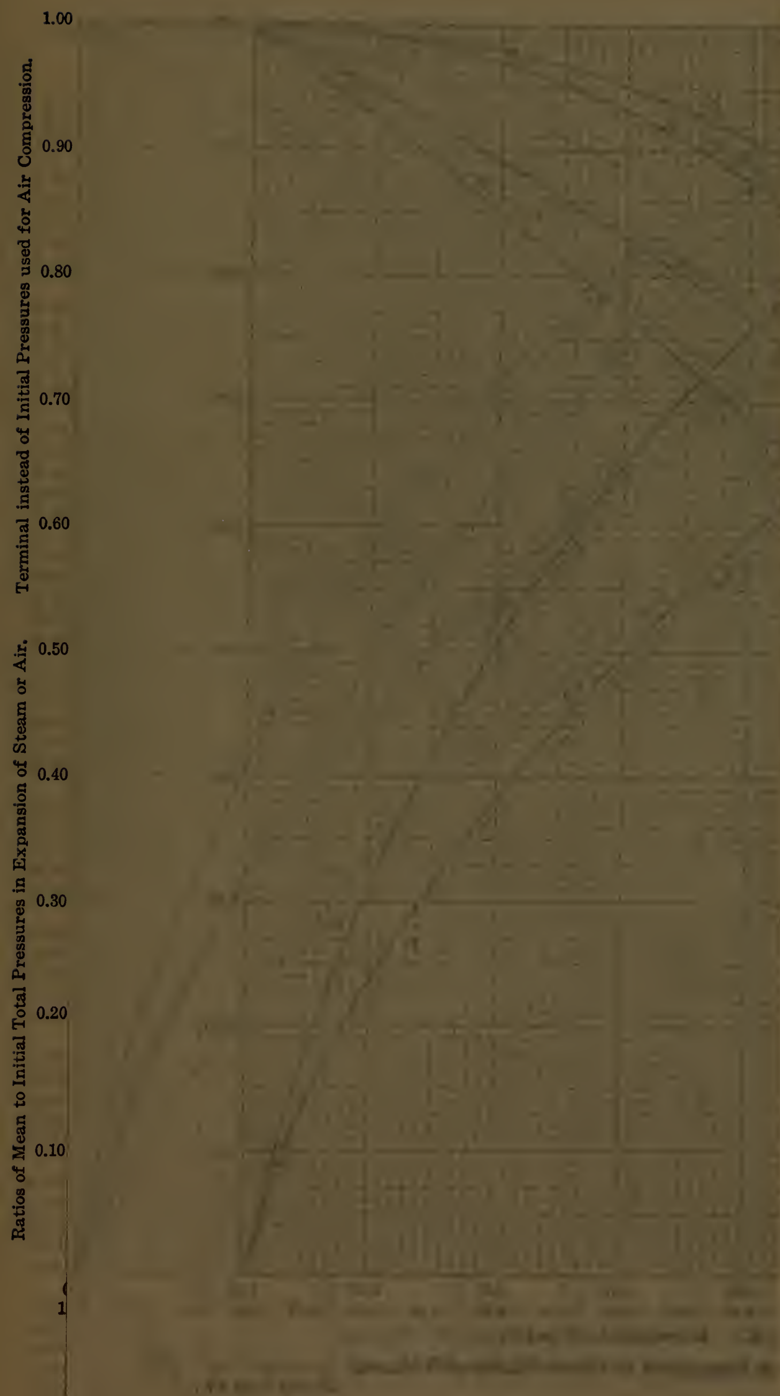
H, Perfect gas, temperature constant.

J, Air expanding (or being compressed) without loss or gain of heat.

K, Saturated steam.

TABLE VIII.—RATIOS AND RECIPROCALS OF MEAN EFFECTIVE PRESSURES FOR GIVEN RATIOS OF VOLUMES

Given ratio	Recip.	H	Recip.	J	Recip.	K	Recip.	L	Recip.	M	Recip.	N	Recip.
0.01	100.0	0.0561	17.825	0.0308	32.467	0.0500	20.000	0.0465	21.505	0.0210	47.619	0.0404	24.752
0.02	50.0	0.0982	10.018	0.0591	16.920	0.0894	11.185	0.0798	12.531	0.0399	25.313	0.0708	14.124
0.03	33.33	0.1352	7.395	0.0860	11.628	0.1245	8.032	0.1085	9.217	0.0577	17.330	0.0974	10.266
0.04	25.00	0.1688	5.924	0.1171	8.952	0.1566	6.385	0.1341	7.457	0.0747	13.387	0.1215	8.230
0.05	20.00	0.1998	5.005	0.1365	7.326	0.1866	5.359	0.1577	6.334	0.0910	10.989	0.1438	6.954
0.06	16.667	0.2288	4.371	0.1604	6.234	0.2148	4.655	0.1796	5.567	0.1068	9.363	0.1647	6.071
0.07	14.286	0.2562	3.903	0.1836	5.446	0.2415	4.140	0.2002	4.995	0.1222	8.183	0.1844	5.423
0.08	12.50	0.2821	3.544	0.2061	4.852	0.2669	3.746	0.2196	4.553	0.1371	7.294	0.2032	4.921
0.09	11.111	0.3067	3.260	0.2280	4.386	0.2912	3.434	0.2382	4.198	0.1517	6.592	0.2211	4.522
0.10	10.00	0.3303	3.027	0.2493	4.011	0.3145	3.179	0.2558	3.909	0.1659	6.027	0.2383	4.280
0.11	9.09	0.3528	2.834	0.2701	3.702	0.3368	2.969	0.2728	3.665	0.1798	5.561	0.2548	3.924
0.12	8.333	0.3744	2.679	0.2903	3.444	0.3583	2.791	0.2891	3.459	0.1935	5.167	0.2708	3.692
0.13	7.692	0.3952	2.533	0.3100	3.258	0.3790	2.638	0.3049	3.279	0.2069	4.833	0.2862	3.494
0.14	7.143	0.4153	2.407	0.3293	3.036	0.3990	2.506	0.3201	3.124	0.2201	4.543	0.3012	3.320
0.15	6.667	0.4346	2.301	0.3481	2.872	0.4183	2.390	0.3348	2.986	0.2331	4.290	0.3157	3.167
0.16	6.25	0.4532	2.206	0.3665	2.728	0.4370	2.288	0.3491	2.864	0.2458	4.068	0.3298	3.032
0.17	5.882	0.4712	2.122	0.3845	2.601	0.4552	2.197	0.3629	2.755	0.2584	3.870	0.3436	2.910
0.18	5.556	0.4887	2.046	0.4020	2.487	0.4727	2.115	0.3764	2.656	0.2708	3.692	0.3570	2.801
0.19	5.263	0.5055	1.978	0.4192	2.385	0.4897	2.042	0.3896	2.567	0.2830	3.533	0.3700	2.702
0.20	5.00	0.5219	1.904	0.4360	2.293	0.5062	1.975	0.4024	2.485	0.2950	3.390	0.3828	2.612
0.21	4.762	0.5377	1.859	0.4524	2.210	0.5223	1.914	0.4149	2.410	0.3069	3.258	0.3953	2.529
0.22	4.545	0.5531	1.808	0.4685	2.134	0.5378	1.859	0.4271	2.341	0.3186	3.138	0.4075	2.454
0.23	4.348	0.5680	1.760	0.4842	2.065	0.5530	1.808	0.4390	2.277	0.3302	3.028	0.4194	2.384
0.24	4.167	0.5825	1.716	0.4996	2.001	0.5677	1.761	0.4507	2.209	0.3416	2.927	0.4312	2.319
0.25	4.00	0.5966	1.676	0.5147	1.942	0.5820	1.718	0.4621	2.164	0.3529	2.833	0.4426	2.259
0.26	3.846	0.6102	1.638	0.5295	1.888	0.5959	1.678	0.4733	2.112	0.3641	2.746	0.4539	2.203
0.27	3.704	0.6235	1.604	0.5439	1.838	0.6094	1.641	0.4843	2.066	0.3752	2.665	0.4650	2.150
0.28	3.571	0.6364	1.571	0.5580	1.792	0.6226	1.606	0.4950	2.020	0.3861	2.590	0.4759	2.101
0.29	3.448	0.6490	1.541	0.5718	1.748	0.6355	1.573	0.5056	1.977	0.3970	2.519	0.4866	2.055
0.30	3.333	0.6612	1.512	0.5854	1.708	0.6479	1.543	0.5160	1.938	0.4077	2.453	0.4971	2.011
0.31	3.226	0.6731	1.485	0.5986	1.670	0.6601	1.515	0.5262	1.900	0.4183	2.391	0.5074	1.970
0.32	3.125	0.6846	1.461	0.6116	1.635	0.6719	1.488	0.5362	1.865	0.4288	2.332	0.5176	1.932
0.33	3.03	0.6959	1.437	0.6243	1.602	0.6835	1.463	0.5461	1.831	0.4393	2.276	0 5276	1.895
0.34	2.941	0.7068	1.414	0.6367	1.570	0.6947	1.439	0.5558	1.799	0.4496	2.224	0.5374	1.861
0.35	2.857	0.7174	1.394	0.6489	1.541	0.7056	1.417	0.5653	1.768	0.4598	2.177	0.5471	1.827
0.36	2.778	0.7278	1.374	0.6608	1.513	0.7163	1.396	0.5747	1.757	0.4700	2.127	0.5567	1.796
0.37	2.703	0.7379	1.355	0.6724	1.487	0.7267	1.376	0.5839	1.712	0.4800	2.083	0.5662	1.766
0.38	2.632	0.7477	1.337	0.6838	1.462	0.7368	1.357	0.5930	1.686	0.4900	2.040	0.5755	1.737
0.39	2.564	0.7572	1.321	0.6949	1.439	0.7466	1.339	0.6020	1.661	0.4999	2.000	0.5846	1.710
0.40	2.50	0.7665	1.304	0.7058	1.416	0.7562	1.322	0.6109	1.636	0.5097	1.961	0.5937	1.684
0.41	2.439	0.7756	1.289	0.7165	1.395	0.7656	1.306	0.6196	1.613	0.5194	1.925	0.6026	1.659
0.42	2.381	0.7844	1.274	0.7269	1.375	0.7747	1.291	0.6282	1.592	0.5291	1.890	0.6115	1.633
0.43	2.326	0.7929	1.261	0.7370	1.356	0.7835	1.276	0.6367	1.570	0.5386	1.856	0.6202	1.612
0.44	2.273	0.8012	1.248	0.7470	1.338	0.7921	1.262	0.6451	1.550	0.5481	1.824	0.6288	1.590
0.45	2.222	0.8093	1.235	0.7567	1.321	0.8005	1.249	0.6533	1.530	0.5576	1.793	.06373	1.569
0.46	2.174	0.9172	1.223	0.7662	1.305	0.8087	1.236	0.6615	1.510	0.5670	1.763	0.6457	1.548
0.47	2.128	0.8249	1.212	0.7754	1.289	0.8166	1.224	0.6696	1.493	0.5762	1.735	0.6540	1.529
0.48	2.083	0.8323	1.201	0.7845	1.274	0.8244	1.213	0.6775	1.476	0.5855	1.708	0.6622	1.510
0.49	2.041	0.8395	1.191	0.7933	1.265	0.8319	1.202	0.6854	1.459	0.5947	1.681	0.6703	1.492
0.50	2.00	0.8466	1.181	0.8019	1.247	0.8392	1.191	0.6932	1.442	0.6038	1.656	0.6784	1.474



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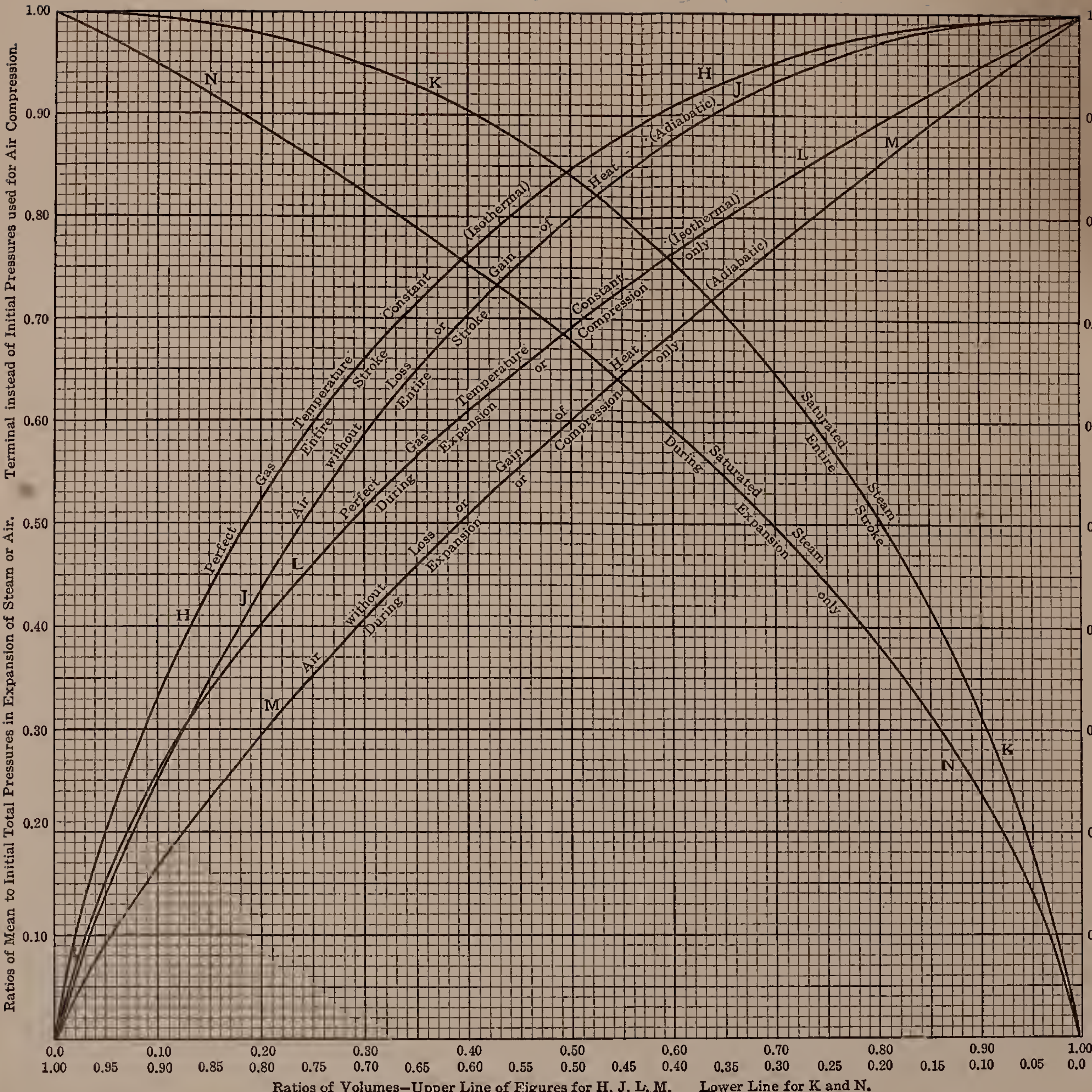

Fig. 7.—Ratios of Mean Effective Pressures in Compression or Expansion to Given Ratios of Volumes.

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TAB

Given ratio	
0.01	10
0.02	5
0.03	3
0.04	2
0.05	2
0.06	1
0.07	1
0.08	1
0.09	11
0.10	1
0.11	
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0.32	
0.33	
0.34	
0.35	
0.36	
0.37	
0.38	
0.39	
0.40	
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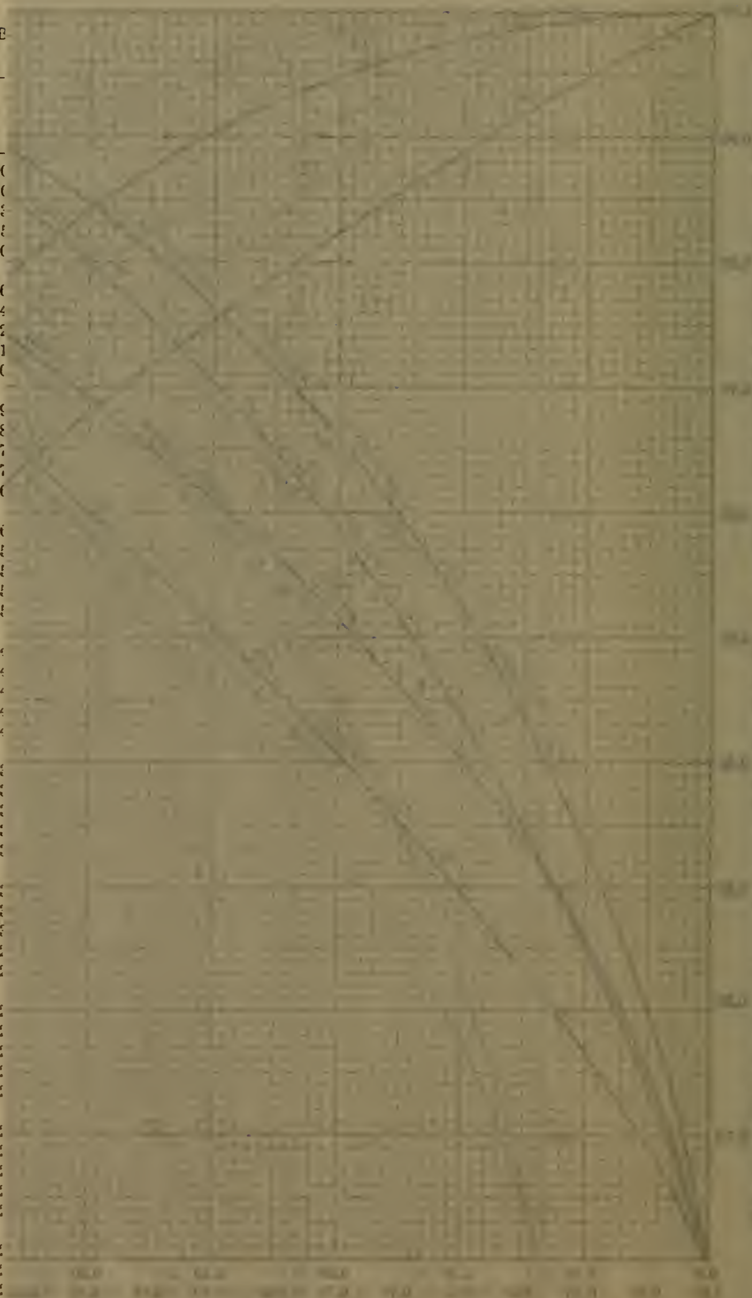

TABLE VIII.—RATIOS AND RECIPROCALS OF MEAN EFFECTIVE PRESSURES FOR GIVEN RATIOS OF VOLUMES—*Continued*

Given ratio.	Recip.	H	Recip.	J	Recip.	K	Recip.	L	Recip.	M	Recip.	N	Recip.
0.51	1.961	0.8534	1.171	0.8103	1.234	0.8463	1.181	0.7008	1.426	0.6128	1.631	0.6863	1.457
0.52	1.923	0.8600	1.163	0.8185	1.222	0.8532	1.172	0.7084	1.410	0.6218	1.608	0.6941	1.441
0.53	1.887	0.8665	1.154	0.8264	1.210	0.8599	1.162	0.7159	1.396	0.6307	1.585	0.7019	1.424
0.54	1.852	0.8727	1.147	0.8342	1.191	0.8664	1.154	0.7234	1.382	0.6396	1.563	0.7096	1.409
0.55	1.818	0.8788	1.137	0.8418	1.188	0.8727	1.145	0.7307	1.368	0.6484	1.542	0.7172	1.394
0.56	1.786	0.8847	1.130	0.8492	1.177	0.8789	1.137	0.7380	1.355	0.6572	1.521	0.7247	1.379
0.57	1.754	0.8904	1.123	0.8563	1.167	0.8848	1.130	0.7451	1.342	0.6659	1.501	0.7322	1.365
0.58	1.724	0.8959	1.116	0.8633	1.158	0.8906	1.122	0.7522	1.329	0.6745	1.482	0.7396	1.352
0.59	1.695	0.9013	1.109	0.8701	1.149	0.8962	1.115	0.7593	1.317	0.6831	1.463	0.7469	1.338
0.60	1.667	0.9065	1.103	0.8767	1.140	0.9017	1.109	0.7662	1.305	0.6917	1.445	0.7541	1.326
0.61	1.639	0.9115	1.097	0.8831	1.130	0.9069	1.102	0.7731	1.293	0.7002	1.428	0.7613	1.313
0.62	1.613	0.9164	1.091	0.8893	1.124	0.9120	1.096	0.7800	1.282	0.7086	1.411	0.7684	1.301
0.63	1.587	0.9211	1.085	0.8953	1.116	0.9169	1.090	0.7867	1.271	0.7170	1.394	0.7755	1.289
0.64	1.563	0.9256	1.080	0.9011	1.109	0.9217	1.085	0.7934	1.259	0.7254	1.378	0.7824	1.278
0.65	1.538	0.9300	1.075	0.9068	1.102	0.9263	1.079	0.8000	1.250	0.7337	1.362	0.7894	1.266
0.66	1.515	0.9342	1.070	0.9123	1.096	0.9307	1.074	0.8066	1.239	0.7419	1.347	0.7962	1.256
0.67	1.493	0.9383	1.065	0.9176	1.089	0.9350	1.069	0.8131	1.229	0.7501	1.333	0.8030	1.245
0.68	1.471	0.9423	1.061	0.9227	1.083	0.9391	1.064	0.8196	1.220	0.7583	1.320	0.8097	1.235.
0.69	1.449	0.9460	1.057	0.9276	1.078	0.9431	1.060	0.8259	1.210	0.7664	1.304	0.8164	1.224
0.70	1.429	0.9497	1.053	0.9324	1.072	0.9469	1.056	0.8322	1.201	0.7745	1.291	0.8230	1.215
0.71	1.408	0.9532	1.049	0.9369	1.067	0.9506	1.052	0.8385	1.192	0.7826	1.277	0.8296	1.205
0.72	1.389	0.9565	1.045	0.9414	1.062	0.9541	1.048	0.8447	1.183	0.7906	1.261	0.8361	1.196
0.73	1.370	0.9597	1.042	0.9456	1.057	0.9575	1.044	0.8509	1.175	0.7885	1.252	0.8426	1.186
0.74	1.357	0.9628	1.038	0.9497	1.053	0.9607	1.040	0.8570	1.166	0.8064	1.240	0.8490	1.177
0.75	1.333	0.9658	1.035	0.9536	1.048	0.9638	1.037.	0.8631	1.158	0.8143	1.228	0.8553	1.169
0.76	1.316	0.9686	1.032	0.9573	1.044	0.9668	1.034	0.8691	1.150	0.8221	1.216	0.8616	1.161
0.77	1.299	0.9713	1.029	0.9609	1.040	0.9696	1.030	0.8750	1.142	0.8300	1.204	0.8679	1.152
0.78	1.282	0.9738	1.026	0.9643	1.036	0.9723	1.028	0.8809	1.135	0.8377	1.193	0.8741	1.144
0.79	1.266	0.9762	1.024	0.9675	1.033	0.9749	1.025	0.8868	1.127	0.8455	1.182	0.8803	1.136
0.80	1.25	0.9785	1.022	0.9706	1.030	0.9773	1.023	0.8926	1.120	0.8532	1.172	0.8864	1.128
0.81	1.235	0.9807	1.020	0.9736	1.027	0.9796	1.021.	0.8983	1.113	0.8608	1.161	0.8925	1.120
0.82	1.22	0.9827	1.017	0.9763	1.024	0.9817	1.018	0.9041	1.106	0.8684	1.151	0.8985	1.113
0.83	1.205	0.9847	1.015	0.9789	1.021	0.9838	1.016	0.9097	1.099	0.8760	1.141	0.9045	1.105
0.84	1.19	0.9865	1.013	0.9814	1.019	0.9857	1.014	0.9154	1.092	0.8836	1.129	0.9104	1.098
0.85	1.176	0.9881	1.012	0.9837	1.016	0.9874	1.012	0.9209	1.085	0.8911	1.122	0.9162	1.091
0.86	1.163	0.9897	1.0104	0.9858	1.014	0.9891	1.010	0.9265	1.079	0.8986	1.112	0.9221	1.084
0.87	1.149	0.9912	1.0089	0.9878	1.012	0.9906	1.009	0.9320	1.072	0.9060	1.103	0.9279	1.077
0.88	1.136	0.9925	1.0075	0.9896	1.010	0.9920	1.008	0.9375	1.066	0.9134	1.094	0.9337	1.071
0.89	1.124	0.9937	1.0061	0.9913	1.0087	0.9933	1.0067	0.9429	1.060	0.9208	1.086	0.9394	1.063
0.90	1.111	0.9948	1.0052	0.9928	1.0072	0.9945	1.0055	0.9482	1.054	0.9282	1.077	0.9451	1.058
0.91	1.099	0.9958	1.0042	0.9942	1.0058	0.9956	1.0044	0.9536	1.048	0.9355	1.069	0.9508	1.051
0.92	1.087	0.9967	1.0033	0.9954	1.0046	0.9965	1.0035	0.9589	1.042	0.9428	1.060	0.9564	1.044
0.93	1.075	0.9975	1.0025	0.9965	1.0035	0.9973	1.0027	0.9642	1.037	0.9500	1.052	0.9620	1.039
0.94	1.064	0.9982	1.0018	0.9974	1.0026	0.9981	1.0019	0.9694	1.031	0.9572	1.044	0.9675	1.033
0.95	1.053	0.9987	1.0013	0.9982	1.0018	0.9987	1.0013	0.9746	1.026	0.9645	1.036	0.9730	1.0277
0.96	1.042	0.9992	1.0008	0.9989	1.0011	0.9991	1.0009	0.9797	1.0207	0.9716	1.029	0.9785	1.0219
0.97	1.031	0.9996	1.0004	0.9994	1.0006	0.9995	1.0005	0.9849	1.0155	0.9788	1.021	0.9840	1.0162
0.98	1.02	0.9998	1.0002	0.9997	1.0003	0.9998	1.0002	0.9899	1.0102	0.9859	1.014	0.9895	1.0106
0.99	1.01	0.9999	1.0001	0.9999	1.0001	0.9999	1.0001	0.9950	1.005	0.9929	1.007	0.9947	1.0053
1.00	1.00	1.00	1.00.	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

DURING EXPANSION OR COMPRESSION ONLY

L, Perfect gas, temperature constant.

M, Air expanding (or being compressed) without loss or gain of heat.

N, Saturated steam.

The following are the formulas by which the computations were made, $\frac{1}{R}$ being the given ratio:

$$H \ldots\ldots \frac{1 + hyp.\ log.\ R}{R}$$

$$J \ldots\ldots \frac{3.451 - 2.451 + \left(\frac{1}{R}\right)^{0.408}}{R}$$

$$K \ldots\ldots 17 \times \frac{1}{R} - 16 \times \left(\frac{1}{R}\right)^{\frac{17}{16}}$$

$$L \ldots\ldots \frac{hyp.\ log.\ R}{R-1}$$

$$M \ldots\ldots \frac{2.451 \times \left[1 - \left(\frac{1}{R}\right)^{0.408}\right]}{\mathrm{R}-1}$$

$$N \ldots\ldots \frac{16 \times \left[1 - \left(\frac{1}{R}\right)^{\frac{1}{16}}\right]}{R-1}$$

Habits of computation vary with the individual, some paying more attention to minute points than others, and some doing mentally what would compel a liberal use of pencil and paper by others. In any case the detailed descriptions of arithmetical operations are apt to make them appear more complicated than they really are.

We will take here a very simple case of single-stage compression. Say that we have a straight-line compressor whose air cylinder is 24-in. diameter and 36-in. stroke, piston rod 4 in., running at 72 r.p.m. and compressing the air to 70 lb., gage; what will be the theoretic horse-power required for adiabatic compression? The area of a 24-in. circle is 452.39 sq. in. and of a 4-in. is 12.56. Then $(452.39+452.39-12.56)\div 2=446$ sq. in., as the mean piston area for both strokes.

As the compression is to 70 lb., gage, the absolute pressure is

70+14.7=84.7, and the pressure ratio is 14.7÷84.7=0.1736. Referring to Table VII, to get precisely the ratio of volumes, using column A, we find that for 0.17, or 0.1700, it is 0.2841, and for 0.1800 it is 0.2959, the difference being 0.2959−0.2841= 0.0118, then 0.0118×0.36=0.004248, and 0.2841+0.0042= 0.2883 as the precise ratio of volumes.

Coming now to the table before us, using column J, we find that for 0.2800 volume ratio, the mean effective pressure ratio is 0.5580 and for 0.2900 it is 0.5718, the difference being 0.0138, then 0.0138×0.83=0.0114 and 0.5580+0.0114=0.5694 as the precise mean effective pressure ratio.

It is usually not necessary, or, indeed worth while, to take all this pains to get the last two figures of the ratio, and in using the diagrams alone the operation would be much simpler. The pressure ratio, 0.1736, we would fix in our minds as 0.17 1/3+, and finding this point on the base-line of the previous diagram, Fig. 6, we would read the vertical distance up to curve A as 0.28 7/8−. Then turning to the diagram now before us, with 0.28 7/8−on the base-line, we would read the vertical distance to curve J as 0.57, which would be sufficiently close to the 0.5694 obtained from the table.

The absolute terminal pressure of compression being 84.7, the absolute mean effective pressure is 84.7×0.5694=48.23, and the working mean effective is 48.23−14.7=33.5.

The theoretic horse-power then is 446, mean piston area,× 33.5, mean effective resistance,×6 ft., per double stroke,×72 r.p.m. ÷33,000 ft.-lb.=196 h.p., to which is to be added for friction whatever percentage information may warrant or judgment suggest.

A HANDY TABLE FOR SIMPLE COMPRESSION COMPUTATIONS

Table IX was produced by the aid of those preceding it. The writer has used it in his own practice for many years, and its value for general use has been attested by its frequent reproduction in pocket books, air machinery builders' catalogues, etc. It is serviceable in many ways in general computations relating to the compression of air and requires little explanation. Throughout the table the air is assumed to be compressed from the normal pressure of 1 atmosphere, 14.7 lb. absolute, and from an initial temperature of 60° F. The first three columns of the

TABLE IX.—VOLUMES, MEAN PRESSURES, TEMPERATURES, ETC., IN THE OPERATION OF AIR COMPRESSION FROM I ATMOSPHERE AND 60° FAHR.

1	2	3	4	5	6	7	8	9	10	11
Gage pressure	Absolute pressure	Pressure in atmospheres	Volume with air at constant temperature.	Volume with air not cooled.	Mean pressure per stroke. Air constant temperature.	Mean pressure per stroke. Air not cooled	Mean pressure during compression only. Air constant temperature	Mean pressure during compression only. Air not cooled.	Final temperatures. Air not cooled.	Gage pressure
0	14.7	1.0	1.0	1.0	0.0	0.0	0.0	0.0	60.0	0
1	15.7	1.068	0.9363	0.95	0.96	0.975	0.43	0.44	71.0	1
2	16.7	1.136	0.8803	0.91	1.87	1.91	0.95	0.96	80.4	2
3	17.7	1.204	0.8305	0.876	2.72	2.8	1.4	1.41	88.9	3
4	18.7	1.272	0.7861	0.84	3.53	3.67	1.84	1.86	98.0	4
5	19.7	1.34	0.7462	0.81	4.3	4.5	2.22	2.26	106.0	5
10	24.7	1.68	0.5952	0.69	7.62	8.27	4.14	4.26	145.0	10
15	29.7	2.02	0.495	0.606	10.33	11.51	5.77	5.99	178.0	15
20	34.7	2.36	0.4237	0.543	12.62	14.4	7.2	7.58	207.0	20
25	39.7	2.7	0.3703	0.494	14.59	17.01	8.49	9.05	234.0	25
30	44.7	3.04	0.3280	0.4638	16.34	19.4	9.66	10.39	255.0	30
35	49.7	3.381	0.2957	0.42	17.92	21.6	10.72	11.59	281.0	35
40	54.7	3.721	0.2687	0.393	19.32	23.66	11.7	12.8	302.0	40
45	59.7	4.061	0.2462	0.37	20.52	25.59	12.62	13.95	321.0	45
50	64.7	4.401	0.2272	0.35	21.79	27.39	13.48	15.05	339.0	50
55	69.7	4.741	0.2109	0.331	22.77	29.11	14.3	15.98	357.0	55
60	74.7	5.081	0.1968	0.3144	23.84	30.75	15.05	16.98	375.0	60
65	79.7	5.423	0.1844	0.301	24.77	32.33	15.76	17.88	389.0	65
70	84.7	5.762	0.1735	0.288	26.0	33.73	16.43	18.74	405.0	70
75	89.7	6.102	0.1639	0.276	26.65	35.23	17.09	19.54	420.0	75
80	94.7	6.442	0.1552	0.267	27.33	36.6	17.7	20.5	432.0	80
85	99.7	6.782	0.1474	0.2566	28.05	37.94	18.3	21.22	447.0	85
90	104.7	7.122	0.1404	0.248	28.78	39.18	18.87	22.9	459.0	90
95	109.7	7.462	0.134	0.24	29.53	40.4	19.4	22.77	472.0	95
100	114.7	7.802	0.1281	0.232	30.07	41.6	19.92	23.43	485.0	100
105	119.7	8.142	0.1228	0.2254	30.81	42.78	20.43	24.17	496.0	105
110	124.7	8.483	0.1178	0.2189	31.39	43.91	20.9	24.85	507.0	110
115	129.7	8.823	0.1133	0.2129	31.98	44.98	21.39	25.54	518.0	115
120	134.7	9.163	0.1091	0.2073	32.54	46.04	21.84	26.2	529.0	120
125	139.7	9.503	0.1052	0.202	33.07	47.06	22.26	26.81	540.0	125
130	144.7	9.843	0.1015	0.1969	33.57	48.1	22.69	27.42	550.0	130
135	149.7	10.183	0.0981	0.1922	34.05	49.1	23.08	28.05	560.0	135
140	154.7	10.523	0.095	0.1878	34.57	50.02	23.41	28.66	570.0	140
145	159.7	10.864	0.0921	0.1837	35.09	51.0	23.97	29.26	580.0	145
150	164.7	11.204	0.0892	0.1796	35.48	51.89	24.28	29.82	589.0	150
160	174.7	11.88	0.0841	0.1722	36.29	53.65	24.97	30.91	607.0	160
170	184.7	12.56	0.0796	0.1657	37.2	55.39	25.71	32.03	624.0	170
180	194.7	13.24	0.0755	0.1595	37.96	57.01	26.36	33.04	640.0	180
190	204.7	13.92	0.0718	0.154	38.68	58.57	27.02	34.06	657.0	190
200	214.7	14.6	0.0685	0.149	39.42	60.14	27.71	35.02	672.0	200

table are of course different forms of the same thing, the pressure to which the air is compressed. The last column of the table is also the same as the first merely for the convenience of following the lines of figures. The first column gives the pressures as they would actually be shown by a steam- or pressure-gage. It would be the actual available working pressure of the air after compression. The second column, or the absolute pressure, is obtained by adding the normal atmospheric pressure, 14.7 lb., to the gage pressure. The third column, showing the pressure in atmospheres, is obtained by dividing the absolute pressure by the normal atmospheric pressure, 14.7 lb.

Column 4 gives the volume of air (the initial volume being 1) after isothermal compression to the given pressure; that is, assuming that the temperature of the air has not been allowed to rise during the compression, or that, if the air has not been completely cooled during the compression, it has been cooled to the initial temperature after the compression. In this case the volume is assumed to be inversely as the absolute pressure, which is very nearly correct. The figures in column 4 are in fact reciprocals of those in column 3, and they are obtained by dividing 1 by the several successive values in column 3. Thus, for a gage pressure of 50 lb., the volume by isothermal compression should be $1 \div 4.401 = 0.2272$, as given in column 4. The compressed volume while in the compressing cylinder, or at the moment of discharge, will always be greater than given in column 4 for the corresponding pressure, because it is impossible to compress air and at the same time abstract all the heat of compression from it. This column does, however, give the volume of air that will be realized if the air is transmitted to some distance from the compressor, or if it is allowed to give up its heat in any way before it is used. Air will be found to lose its heat very rapidly, and this column may be taken to represent the volume of air after compression actually available for the purpose for which the air may have been compressed.

Column 5 of the table gives the volume of air at the completion of the compression, assuming that the air has neither lost nor gained any heat during the compression, and that all the heat developed by the compression remains in the air. This column shows the air more nearly as the compressor actually has to deal with it. In any compressor the air will lose some of its heat during the compression, and the air is never as hot during the compression nor at the completion of the compression as theory says it should be. The theory is all right but the air does lose some of its heat, as is evidenced by the heating of the cylinder and the necessity of water-jacketing. The slower the compressor runs, within reasonable limits and other things being equal, the better chance the air has to give up some of its heat, consequently the smaller will be its volume all through the operation, and the less will be the power required for the compression.

Column 6 gives the mean effective resistance to be overcome by the air-cylinder piston in the stroke of compression, assuming that the air throughout the operation remains constantly at its initial temperature—isothermal compression. Of course the air never will remain at constant temperature during compression, and this column remains the ideal to be kept in view and striven for but never more than approximated in practical operation. Column 7 gives the mean effective resistance to be overcome by the piston for the compression stroke, supposing that there is no cooling of the air during the compression—adiabatic compression. As we have seen, there is more or less—generally less, but always some—cooling of the air during its compression, so that the actual mean effective resistance will always be somewhat less than as given in this column; but for computing the actual power required for operating air-compressor cylinders the figures in this column for the given terminal pressures may be taken and a certain percentage added for friction—say 5 per cent.—and the result will represent very closely the power required by the compressor. In proposing to add 5 per cent. for friction we do not mean that the total friction of a steam-actuated air-compressor will be only 5 per cent., for it will probably be more than 10 per cent., but part of this 10 per cent. will have been conpensated for by the partial cooling of the air during the compression.

The values given in columns 6 and 7 are used in computing the horse-power of an air-compressing cylinder precisely as the mean effective pressure per stroke in a steam-cylinder is used in computing its power. In the steam-cylinder the computation gives the power developed by the steam, and the same system of computation applied to the air-cylinder gives the power used in the compression.

Having an air-compressing cylinder 20 in. diameter × 2 ft. stroke at 75 r.p.m., or 300 ft. piston speed, compressing air adiabatically to 75 lb., we find in column 7 that the mean effective pressure is 35.23 lb., and then the horse-power required will be computed as follows:

$$20^2 \times 0.7854 \times 35.23 \times 300 \div 33,000 = 100 \text{ h.p.}$$

The mean effective pressures given in columns 6 and 7 being for compression to different gage pressures from an initial pressure of 1 atmosphere, it does not follow that those values will be correct for computations in compound compression, or for compression from any other initial pressure but that of 1 atmosphere. Thus in column 7 the M.E.P. for compressing from 1 atmosphere to 50 lb. gage pressure is 27.39. In this case the pressure of the air compressed is increased 50 lb., but it does not follow that we can take air at 50 lb. and compress it to 100 lb. with the same mean effective pressure. In the latter case the M.E.P. required would be 40.33, or 47 per cent. greater than in the former case.

Column 8 gives the mean effective resistance for the compression part only of the stroke in compressing air isothermally from a pressure of 1 atmosphere to any given pressure. This at once calls our attention to the two distinct operations involved in practical air-compression: the actual compression of the air to the given pressure, and the delivery or expulsion of the air from the cylinder after the full pressure is attained. These two operations correspond inversely to the two operations occurring in the cylinder of a steam-engine: the admission of the steam, where it is sustained at approximately full pressure until the point of cut off, and the expansion of the steam from the point of cut off to the termination of the stroke, the expansion period in the steam-cylinder corresponding inversely with the compression in the air-cylinder, and the admission of the steam corresponding with the delivery of the air.

It will be noticed that the mean effective pressures in columns 8 and 9, for the compression part only of the stroke, are much lower than those in columns 6 and 7 for the whole stroke, but when to the work of the compression part of the stroke is added the work of delivery, the values will be found to correspond very nearly. Thus when compressing adiabatically to 50 lb. gage pressure the volume of air delivered will be (column 5) 0.35 of the original volume, or 0.35 of the stroke for each cylinderful of free air, so that the pressure or resistance for 0.35 of the stroke will be 50 lb., while for the compression part of the stroke, 1 −

0.35=0.65, the resistance will be 15.05, as given in column 9. Then $(15.05 \times 0.65)+(50 \times 0.35)=27.28$, which corresponds as well as could be expected with the value in column 7 for the whole stroke, 27.39.

There is also to be observed a less proportional difference between the values in columns 8 and 9 than between those in columns 6 and 7, but this also will be found to be compensated for by the differences in terminal volume for isothermal or for adiabatic compression and the different proportion of the stroke occupied by the full pressure of delivery. Thus comparing the figures for isothermal compression with those just given for adiabatic compression, compressing to 50 lb., as before, we have: $(13.48 \times 0.7728)+(50 \times 0.2272)=21.78$, a result which may be said to be identical with the value 21.79 for the whole stroke, as given in column 6.

Columns 8 and 9 may be found serviceable in some cases in computing the power used in the first stage of compound compression, where generally the entire function of the first cylinder is that of compression only, its total contents from the beginning to the end of the stroke being simply compressed into the volume contained in the smaller cylinder, and there being no part of the stroke properly occupied in delivery or expulsion at any completed pressure.

Column 10 gives the theoretical temperature of the air after compression, adiabatic, to the given pressure. As we have remarked elsewhere, the actually observed temperature in these cases is never as high as the theoretical temperature. This is not that the theory is incorrect, for, as usual, the theory is more nearly correct than "practical" people are wont to allow. If the temperature of the compressed air by observation is not found to correspond with the figures as given, it is only because the air is being cooled by conduction or radiation even while it is being heated by compression.

CHAPTER V

THE INDICATOR ON THE AIR-COMPRESSOR

The steam-engine indicator, so called from the incident of its inception, is just as much an air-compressor indicator, and it can tell us just as much, or perhaps a little more, of what goes on in the air-cylinder as it tells us of the steam-cylinder. All the conditions seem specially to invite the application of the indicator to the air-compressor, and to the study of air-compression practice and results by its aid. In fact the air-compressor seems to be the ideal and only perfect field for the indicator. A steam actuated air-compressor may be said to be the only machine where an indicator can be applied and be made to tell the whole story of the power developed and of the work actually done.

In the steam pump of any type the report from the card of the water cylinder is affected by questions relating to the inertia of the body of water. With a steam-engine and the bare testimony of the indicator-card, there is always some uncertainty about the friction of the working parts of the engine. We may take what we are pleased to call the "friction diagram," when the engine is running without doing any external work, and we know what resistance the steam has to overcome at that time; but that tells us comparatively little of the resistance of the engine parts when loaded. We know that the friction of nearly every working part of the engine increases with the load, but, when the load is on, we do not know from the indicator-card how much of its mean effective indicates actual work done, or how much of it belongs to the friction of the engine, and to get the result with any certainty and accuracy it is necessary to employ some form of dynamometer in connection with the indicator, and let them fight it out between them.

In the case of the air-compressor this is all different. The air-compressor is its own dynamometer. By taking cards from both the air- and the steam-cylinders at the same time, or when the compressor is running under the same conditions, we get a perfect statement of the power developed and of the actual work done, and then we know, too, that the difference in indicated horse-power between the air- and the steam-cards clearly shows the power that has been expended merely to keep the machine in motion. The cards not only give the comparative total power and work, but also the relations of the one to the other at any point of the stroke, showing the air resistance at any point, as well as the force of the steam at the same point, and through this knowledge it will advise us whether the air is compressed with economy or whether better results are to be sought for.

Realizing the importance of the indicator as an indispensable aid in the full development of economical air-compression, it is proper that we learn what we can of the peculiarities of the air-card and of the means of manipulating and interpreting it. We can only consider at first the card from the single air-cylinder, in which the whole operation of air-compression is completed at a single stroke. The cards from cylinders in which either stage of a compound compression is carried on assume peculiar shapes, which we may find pleasure in studying later on.

To an indicator-man who has been brought up, as most have, exclusively upon steam-cards the air-card is at first a little confusing, from the fact that all the operations upon the one card are the reverse of those upon the other. The admission-line of the steam-card is the delivery-line of the air-card; the expansion-line in the one is the compression-line in the other; the exhaust or back-pressure line is the admission-line, and the compression-line becomes the re-expansion-line. One can, however, soon "catch on" and become familiar with each operation and the way it is shown by the lines of the diagram.

It is not the purpose of this work to instruct in the application and use of the indicator. We must assume that it is in competent hands, or its evidence will be worthless. Indicator-cards have, however, a way of telling for themselves frequently if they have not been taken with a reasonable regard for the essential conditions. As the peculiarly important part of the air-card is the compression-line, it is necessary that the drum movement be correct, and that, in proportion to its length, the travel of the card shall be accurately coincident with the piston travel at all points. Cards, to be relied upon, should not be taken until the compressor has been run long enough to have attained its complete working conditions. We know that the compression of air heats it, and that the heat then in the air is communicated more or less to everything in contact with it. When the cylinder becomes heated, it has its effect back again upon the air, and until the compressor has been run continuously and at full pressure for an hour or so, the full temperature of the working parts has hardly been reached, and the effect of the heated parts upon the temperature of the air at different points of the stroke will not be correctly indicated. Cards taken from a compressor that has only just been started will give a lower compression-

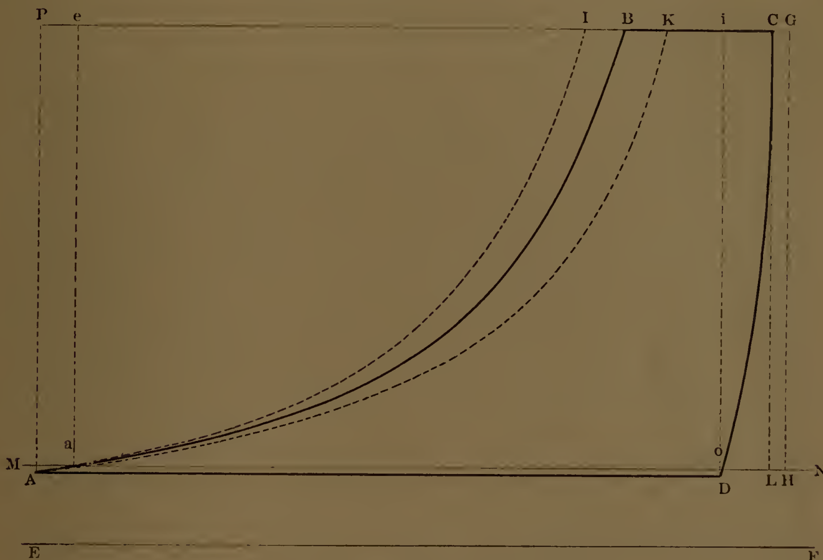

FIG. 8.—Theoretical Indicator Card.

line and a lower mean effective pressure (M. E. P.) than those taken after the cylinder and piston and connecting parts have been heated up to their mean working temperature.

Fig. 8 is offered as an ideal and typical single-compression air-cylinder card, designed to show the points and properties of the card, and the methods of manipulating and studying it. The card is somewhat smoother and cleaner and in most respects more perfect than any actual card, except that the admission-line is purposely drawn rather low to keep it perfectly distinct from the atmosphere-line. The lines constituting the actual diagram are as follows:

AB, Compression-line
BC, Delivery-line
CD, Re-expansion-line
DA, Admission-line

These constitute the actual card, and together represent the complete cycle of operations occurring in one end of the air-cylinder for one complete revolution of the compressor-crank. The atmosphere-line, *MN*, also is traced by the indicator, and is the neutral line of the diagram, or the line of departure in air-compression.

For the proper interpretation of the diagram additional lines are to be drawn as follows: *EF*, the line of perfect vacuum. This line is drawn parallel to the atmosphere-line, *MN*, and at a distance below it determined by the scale of the diagram. The pressure of the atmosphere at sea-level being 14.7, and always decreasing as the altitude increases, the practice of calling the atmospheric pressure 15 lb. may be said to be a rather loose one. If the compressor is operated at a considerable altitude above the sea-level, as many are, the atmospheric pressure at the time and place where the diagram is taken should be ascertained by a barometer, and the line *EF* be drawn accordingly. It should be remembered, as we will see when we get to it, that a height of only a quarter of a mile, or a little over 1300 ft. will make a difference of 7 per cent. in the volume of air delivered.

The vertical lines *PA* and *CL* having been drawn perpendicular to *MN*, defining the extreme length of the actual diagram, the clearance-line *GH* may next be drawn. This is drawn parallel to *CL*, and the distance *CG* or *LH* may be ascertained by computation as follows: The volume represented by the rectangle *APCL* is the actual displacement of the piston for its whole travel. The volume of air acted upon by the piston is this volume increased by the volume *CGHL* remaining in the clearance-space of the cylinder. This volume of air, *CGHL*, at the end of the compression-stroke, and at the pressure indicated by the diagram, has upon the return stroke of the piston re-expanded until it reached the atmospheric pressure again at *D*. This re-expansion is so quickly accomplished that whatever the temperature at the beginning the re-expansion is practically adiabatic. The relative volume before and after the re-expansion may be found in column 5 of Table VII. Assuming the scale of the diagram to be 30 and the pressure at *CG* to be 70 lb. gage, and designating *LH* by x, we have the proportion

$$x : DL + x :: 0.288 : 1$$

Then the length *DL* being 0.25, in. we have

$$x : 0.25 + x : : 0.288 : 1;$$

then

$$x = 0.072 + 0.288x,$$

and

$$0.712x = 0.072,$$
$$x = 0.101.$$

So that *CG* or *LH* equals say $\frac{1}{10}$ in., and *GH* may be drawn accordingly.

Having drawn *GH*, the rectangle *APGH* represents the total volume of air subjected to compression for the stroke, and noting the point *a*, at which the compression-line begins to rise from the atmosphere-line, and drawing the perpendicular *ae*, then *aeGH* represents the total volume of air at atmospheric pressure. The point *a*, being the point at which compression from atmospheric pressure begins, may be considered the beginning of the whole diagram, and the cycle of operations for the entire stroke may be considered to start from this point.

For computing the mean effective resistance the entire enclosed area of the actual diagram *ABCD* is to be taken, and this area may be measured by the planimeter, or by the mean of a series of ordinates in the customary way, as with any other diagram. The area lying below the atmosphere-line of course represents the resistance upon the return stroke, but the diagrams from both ends of the cylinder being assumed to be similar, the entire area may be taken for the single stroke. The correct practice is to take diagrams from both ends of the cylinder, and it should be followed if possible, but it is clearer and simpler for us here to consider only the single diagram.

The M.E.P. of the diagram having been ascertained, the indicated horse-power (i.h.p.) represented may be computed precisely as in the case of a steam-engine. Thus the M.E.P. in the diagram before us happening to be 30, if it were taken from a cylinder 20 in. diameter × 24 in. stroke at 80 r.p.m. the i.h.p. for the double stroke will be as follows:

$$20^2 \times 0.7854 \times 30 \times 4 \times 80 \div 33{,}000.$$

I like always in such cases to put it down in this way, that I may be sure that I get in all the ingredients. It is not necessary to run for a table of squares or of areas, and no time is saved by doing so. The decimal 0.7854 is always cleanly divisible by the constant divisor 33,000, giving us 0.0000238 as a substitute constant for both of these combined. It is not difficult to remember this or to keep it posted with other labor-saving devices in a convenient place. The ciphers in the other factors will help us to elbow the decimal point to the right, and our case will then stand like this, a little string of simple and easy multiplications:

$$20^2 \times 0.0000238 \times 30 \times 4 \times 80 =$$
$$2^2 \times 0.238 \times 3 \times 4 \times 8 =$$
$$0.238 \times 384 = 91.39 \text{ i.h.p.}$$

We will not here go into the question of the additions to be made to this for friction, etc.

The i.h.p. having been ascertained, that gives us the power consumed, or the cost of the compression, and then we naturally want to know the actual quantity of air compressed and delivered. The indicator-diagram shows this very accurately. At the point *a*, where the compression-line takes its departure from the atmosphere-line, the cylinder is shown to be full of air at the atmospheric pressure and corresponding density. This is not the whole cylinder, as a portion of it, *Aa*, has been already traversed by the piston. Whatever proportional distance the point *a* may be from the beginning of the stroke is to be deducted from the total length of the stroke, and the remainder represents the total actual volume of air at atmospheric pressure subjected to compression for that stroke. The compression and delivery of the air goes on with the advance of the piston until it reaches the extreme end of its stroke at *CL*, but when that is reached, the clearance-space *LCGH* is filled with air compressed, but not delivered, and upon the return of the piston this air re-expands until it reaches the atmosphere-line at *o*, so that practically the travel of the piston from *o* to *L* and back again has accomplished nothing toward compression, and the distance *oL* also is to be deducted from the total length of the line *AL*, when that line is taken to represent the volume of air compressed and delivered. In the diagram before us, if *AL* be $3\frac{7}{8}$ in. and *ao* be $3\frac{7}{16}$ in., the ratio of air compressed and delivered is

$\frac{3.4375}{3.875}=88$ per cent. of the cylinder capacity. As was remarked, this card does not represent actual practice, and the ratio is generally not as low as this, being seldom less than 5 per cent. and from that up to 10 per cent. instead of the 12 per cent. here shown.

In compressors having positively moved inlet valves, if the valve is set to open while the crank is on the center, the compressed air in the clearance space is released at once and there is no normal re-expansion-line but a sudden vertical drop from the top, and there is no means of ascertaining the amount of clearance from the card.

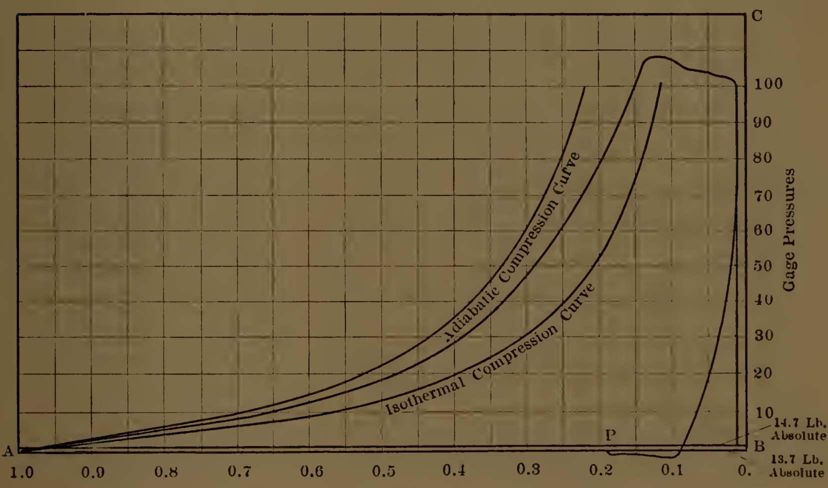

FIG. 9.—Laying Out Adiabatic and Isothermal Curves.

So far as the indicator has anything to say about the economy of the air compression in any given case its evidence is found chiefly in the compression-line of the diagram, and for comparison it is necessary to describe upon the diagram the theoretical isothermal and adiabatic compression curves.

There are various ways of doing this and among the best within my knowledge is that devised by Mr. H. V. Conrad first printed in *Power* and here reproduced in his own words.

"Indicator diagrams taken from air cylinders always show the compression curve as starting below the atmospheric line, when the compressor is drawing free air. This starting-point of compression may range from 1/4 lb., in the high-class machine, to 1 1/2 lb. or more, below the

TABLE X.—DATA FOR LAYING OUT THE THEORETICAL ISOTHERMAL COMPRESSION LINE

Absolute initial pressure, pounds	Gage pressure in pounds													
	2.5	5	10	20	30	40	50	60	70	80	90	100	110	130	150
14.7	0.855	0.746	0.595	0.424	0.329	0.269	0.227	0.197	0.174	0.155	0.140	0.128	0.118	0.1016	0.0894
14.6	0.854	0.745	0.593+	0.422	0.327	0.2675	0.226	0.196	0.1728	0.1542	0.1393	0.1273	0.1172	0.101	0.0888
14.5	0.852+	0.743+	0.592+	0.420	0.326	0.266	0.225	0.195	0.1716	0.1535	0.1386	0.1265	0.1165	0.1002	0.0882
14.4	0.852	0.743	0.591	0.418	0.3245	0.2648	0.224	0.1937	0.1706	0.1525	0.1380	0.1258	0.1157	0.0996	0.0876
14.3	0.852−	0.742	0.589	0.417	0.323	0.2635	0.2225	0.1925	0.1696	0.1516	0.1372	0.1250	0.1150	0.0991	0.0870
14.2	0.851	0.741	0.587	0.416	0.3215	0.2622	0.221	0.1915	0.1686	0.1508	0.1364	0.1242	0.1143	0.0986	0.0864
14.1	0.850	0.740	0.585	0.414	0.320	0.261	0.220	0.1905	0.1676	0.1500	0.1356	0.1235	0.1137	0.0979	0.0859
14.0	0.849	0.738	0.583	0.412	0.3185	0.2595	0.219	0.1895	0.1666	0.1491	0.1347	0.1228	0.1130	0.0972	0.0853
13.9	0.848	0.736	0.582	0.411	0.3165	0.2578	0.2175	0.1884	0.1657	0.1482	0.1338	0.1220	0.1123	0.0966	0.0848
13.8	0.847	0.734	0.580	0.409	0.3150	0.2563	0.2165	0.1873	0.1648	0.1472	0.1330	0.1212	0.1116	0.0960	0.0842
13.7	0.846	0.733	0.578	0.407	0.3135	0.2550	0.2152	0.1862	0.1638	0.1462	0.1322	0.1205	0.1109	0.0953	0.0837
13.6	0.845	0.732	0.577	0.405	0.3120	0.2537	0.2140	0.1850	0.1627	0.1453	0.1313	0.1197	0.1101	0.0947	0.0831
13.5	0.844	0.730	0.575	0.403	0.3105	0.2522	0.2125	0.1838	0.1616	0.1444	0.1305	0.1189	0.1093	0.0940	0.0825
13.4	0.843	0.728	0.573	0.402	0.309	0.2510	0.2112	0.1826	0.1606	0.1435	0.1296	0.1181	0.1085	0.0934	0.0820
13.3	0.842	0.726	0.571	0.400	0.307	0.2495	0.2100	0.1814	0.1596	0.1426	0.1286	0.1173	0.1078	0.0928	0.0814
13.2	0.841	0.725	0.569	0.398	0.305	0.248	0.2090	0.1803	0.1586	0.1418	0.1278	0.1166	0.1071	0.0922	0.0809
13.1	0.840	0.724	0.568	0.396	0.304	0.2465	0.2076	0.1792	0.1575	0.1408	0.1270	0.1158	0.1063	0.0916	0.0804
13.0	0.839	0.723	0.566	0.394	0.302	0.2452	0.2062	0.1780	0.1565	0.1398	0.1263	0.1151	0.1057	0.0910	0.0798
12.9	0.838	0.721	0.564	0.3923	0.301	0.2437	0.205	0.1770	0.1556	0.1389	0.1254	0.1142	0.1050	0.0903	0.0792
12.8	0.837	0.719	0.562	0.3908	0.2992	0.2424	0.2035	0.1758	0.1546	0.1379	0.1245	0.1136	0.1043	0.0896	0.0786
12.7	0.836	0.7175	0.560	0.3892	0.2975	0.2410	0.2023	0.1747	0.1536	0.1370	0.1238	0.1128	0.1036	0.0890	0.0781
12.6	0.835	0.716	0.558	0.3875	0.2960	0.2395	0.2012	0.1735	0.1526	0.1361	0.1229	0.1119	0.1028	0.0884	0.0775
12.5	0.834	0.714	0.556	0.3850	0.2942	0.2380	0.2000	0.1725	0.1516	0.1352	0.1220	0.1111	0.1021	0.0877	0.0769
12.4	0.832	0.713	0.554	0.3827	0.2925	0.2368	0.1987	0.1712	0.1505	0.1341	0.121	0.1102	0.1012	0.0871	0.0764
12.3	0.831	0.712	0.552	0.381	0.291	0.2355	0.1974	0.1701	0.1495	0.1331	0.120	0.1094	0.1005	0.0865	0.0758

TABLE X.—DATA FOR LAYING OUT THE THEORETICAL ISOTHERMAL COMPRESSION LINE.—*Continued*

Absolute initial pressure, pounds	Gage pressures in pounds													
	2.5	5	10	20	30	40	50	60	70	80	90	100	110	130	150
12.2	0.830	0.71–	0.550	0.379	0.289	0.2338	0.1961	0.169	0.1485	0.1322	0.1192	0.1086	0.0998	0.0858	0.0752
12.1	0.829	0.709	0.548	0.377	0.2872	0.2321	0.1948	0.1679	0.1474	0.1314	0.1185	0.1078	0.0992	0.0852	0.0747
12.0	0.828	0.707	0.546	0.3755	0.2857	0.2306	0.1936	0.1667	0.1463	0.1306	0.1177	0.1070	0.0985	0.0847	0.0741
11.9	0.827	0.705	0.544	0.3735	0.2842	0.2292	0.1925	0.1656	0.1452	0.1295	0.1168	0.1063	0.0977	0.0840	0.0736
11.8	0.826	0.703	0.542	0.3715	0.2821	0.2278	0.1910	0.1644	0.1441	0.1285	0.1159	0.1055	0.0970	0.0833	0.0730
11.7	0.8245	0.701	0.540	0.3692	0.2805	0.2262	0.1895	0.1632	0.1431	0.1276	0.115	0.0048	0.0963	0.0826	0.0725
11.6	0.823	0.699	0.538	0.367	0.279	0.255	0.1884	0.162	0.142	0.1266	0.1142	0.104	0.0956	0.082	0.0719
11.5	0.8215	0.697	0.536	0.365	0.277	0.2235	0.1872	0.1609	0.141	0.1258	0.1133	0.1032	0.0947	0.0813	0.0713
11.4	0.820	0.695	0.532	0.363	0.2755	0.222	0.186	0.1598	0.140	0.1249	0.1124	0.1024	0.0939	0.0807	0.0707
11.3	0.8185	0.693	0.530+	0.361	0.2738	0.2205	0.1845	0.1585	0.139	0.1239	0.1116	0.1015	0.0932	0.080	0.0701
11.2	0.817	0.6915	0.529–	0.359	0.272	0.219	0.183	0.1574	0.138	0.1229	0.1108	0.1007	0.0925	0.0794	0.0695
11.1	0.8157	0.690	0.527	0.357	0.2704	0.2175	0.1818	0.1564	0.137	0.1219	0.1099	0.0999	0.0917	0.0787	0.0689
11.0	0.8143	0.688	0.5245	0.355	0.2685	0.216	0.1805	0.155	0.136	0.121	0.1090	0.0992	0.091	0.0781	0.0684
10.9	0.8136	0.686	0.522	0.353	0.2665	0.2142	0.179	0.1539	0.1348	0.120	0.1080	0.0984	0.0902	0.0774	0.0678
10.8	0.813	0.684	0.520	0.351	0.2645	0.213	0.1775	0.1525	0.1335	0.119	0.170	0.0976	0.0895	0.0768	0.0672
10.7	0.812	0.682	0.518	0.349	0.263	0.211	0.176	0.1513	0.1325	0.118	0.1061	0.0968	0.0887	0.0761	0.0666
10.6	0.810+	0.680	0.515	0.346+	0.261	0.209	0.175–	0.150	0.1315	0.117	0.1053	0.096	0.088	0.0755	0.0661
10.5	0.808+	0.678	0.512	0.344	0.259	0.208	0.1735	0.149	0.1305	0.116	0.1044	0.0951	0.0872	0.0748	0.0655
10.4	0.807	0.676	0.510	0.342	0.257+	0.2065	0.172	0.148	0.1295	0.115	0.1035	0.0943	0.0864	0.0741	0.0649
10.3	0.805	0.674	0.508	0.340	0.256–	0.205	0.1705	0.1465	0.1283	0.114	0.1026	0.0935	0.0857	0.0735	0.0643
10.2	0.804	0.672	0.506	0.338–	0.254	0.203	0.1695	0.145+	0.1272	0.113	0.1017	0.0926	0.0849	0.0728	0.0638
10.1	0.802	0.669	0.503	0.335	0.252	0.202	0.168+	0.144	0.1261	0.112	0.1008	0.0918	0.0841	0.0721	0.0632
10.0	0.80	0.666	0.50	0.333	0.25	0.20	0.1666	0.143	0.125	0.111	0.10	0.090	0.0834	0.0715	0.0625

TABLE XI.—DATA FOR LAYING OUT THE THEORETICAL ADIABATIC COMPRESSION LINE

Absolute initial pressure, pounds	Gage pressure in pounds													
	2.5	5	10	20	30	40	50	60	70	80	90	100	110	130	150
14.7	0.895	0.812	0.692	0.543	0.454	0.393	0.349	0.315	0.288	0.266	0.248	0.233	0.2192	0.1972	0.180
14.6	0.895−	0.811+	0.691	0.542	0.453	0.392	0.348	0.314	0.287+	0.265	0.247	0.232	0.2181	0.1961	0.179
14.5	0.894+	0.811	0.690	0.541	0.452	0.391	0.347	0.313	0.287−	0.264	0.246+	0.231+	0.217	0.195	0.178
14.4	0.894	0.810	0.688	0.539	0.450	0.389+	0.345+	0.312−	0.285	0.263+	0.245	0.229+	0.216	0.194+	0.177
14.3	0.893	0.809	0.686	0.538	0.488	0.388	0.344	0.310+	0.284	0.262	0.244	0.228+	0.215+	0.193+	0.176+
14.2	0.892	0.808	0.685	0.537	0.447	0.387−	0.342+	0.309	0.283−	0.261−	0.243	0.227+	0.214+	0.193−	0.176−
14.1	0.891	0.807	0.683+	0.535	0.445	0.385+	0.341+	0.308	0.281+	0.260	0.242	0.226+	0.213+	0.192	0.175+
14.0	0.890	0.806	0.682	0.533	0.444−	0.384−	0.340	0.307	0.280+	0.259	0.241	0.226−	0.212+	0.191+	0.175−
13.9	0.889	0.805−	0.681	0.532	0.442	0.382+	0.338	0.305	0.279+	0.258−	0.240	0.225	0.211+	0.190+	0.173+
13.8	0.889−	0.803	0.680−	0.530	0.441	0.381−	0.337	0.304	0.278	0.257−	0.239	0.224−	0.211−	0.189+	0.172+
13.7	0.888	0.802+	0.678	0.528	0.439+	0.379+	0.366	0.303+	0.276+	0.255+	0.238	0.222+	0.210−	0.188+	0.171+
13.6	0.888−	0.801	0.677	0.527	0.437+	0.378−	0.334+	0.302−	0.275+	0.254	0.237	0.221+	0.209−	0.187+	0.171−
13.5	0.877	0.800+	0.676−	0.525	0.436	0.376+	0.333+	0.300+	0.274+	0.253−	0.235+	0.221−	0.208−	0.187−	0.170
13.4	0.887−	0.779	0.674	0.523+	0.434	0.375	0.332−	0.299	0.273+	0.251+	0.234+	0.220−	0.207−	0.186	0.169+
13.3	0.885+	0.798−	0.672	0.522	0.433	0.373+	0.330	0.298−	0.272	0.250+	0.233	0.219−	0.206−	0.185−	0.168−
13.2	0.884	0.796+	0.671	0.520+	0.431+	0.371+	0.329	0.296+	0.270+	0.249+	0.232	0.217+	0.204+	0.184−	0.168−
13.1	0.884−	0.795	0.670−	0.518	0.429	0.369	0.328	0.295	0.269	0.248+	0.231	0.216+	0.203+	0.183	0.167+
13.0	0.883−	0.794	0.667+	0.516	0.428−	0.368	0.327	0.293+	0.268	0.248−	0.230	0.215+	0.202+	0.182	0.166+
12.9	0.882	0.793	0.665+	0.515−	0.426+	0.367+	0.325	0.292+	0.267	0.246+	0.228+	0.214	0.202−	0.181	0.165
12.8	0.881+	0.792	0.664	0.513	0.425−	0.366	0.323	0.291	0.266−	0.245	0.227+	0.213	0.201	0.180+	0.164+
12.7	0.880+	0.791	0.662+	0.512−	0.423	0.364	0.322−	0.290−	0.264	0.244	0.226+	0.212	0.200	0.180−	0.163+
12.6	0.880	0.790−	0.661	0.510	0.421+	0.362+	0.320+	0.288+	0.263	0.243−	0.225+	0.211	0.199	0.179−	0.163−
12.5	0.880−	0.788	0.660−	0.508	0.420	0.361	0.319	0.287	0.262	0.241−	0.224+	0.210	0.198	0.178−	0.162−
12.4	0.878+	0.787−	0.658	0.506	0.418	0.360−	0.318	0.286	0.2605	0.240	0.2235	0.209	0.1967	0.177−	0.161
12.3	0.877	0.785+	0.656	0.504	0.416	0.358	0.317	0.2845	0.259	0.239	0.222	0.208	0.1955	0.176	0.160

TABLE XI.—DATA FOR LAYING OUT THE THEORETICAL ADIABATIC COMPRESSION LINE.—*Continued*

Absolute initial pressure, pounds	Gage pressure in pounds													
	2.5	5	10	20	30	40	50	60	70	80	90	100	110	130	150
12.2	0.876	0.784	0.654	0.502	0.414+	0.3556	0.315	0.283	0.258−	0.238	0.221	0.207−	0.1945	0.175	0.159
12.1	0.8755	0.783	0.652+	0.500	0.413−	0.354	0.313+	0.281+	0.2565	0.237	0.220	0.2055	0.1938	0.174	0.1582
12.0	0.875	0.782	0.651	0.498	0.411	0.353	0.312−	0.280	0.255+	0.236	0.219	0.2046	0.1930	0.173	0.1575
11.9	0.874	0.780+	0.649	0.496	0.409	0.351+	0.310	0.279	0.254	0.234+	0.218	0.2038	0.1920	0.172	0.1567
11.8	0.873	0.779−	0.647	0.4945	0.407	0.350	0.309−	0.278−	0.2527	0.233	0.2167	0.2029	0.1910	0.1712	0.1559
11.7	0.872	0.777	0.646−	0.493	0.405	0.348+	0.307	0.2765	0.2515	0.232−	0.2155	0.2019	0.1900	0.1703	0.1551
11.6	0.871	0.776−	0.644	0.491	0.4035	0.346+	0.3055	0.275	0.2503	0.2307	0.2144	0.2008	0.1888	0.1695	0.1543
11.5	0.870−	0.774	0.642	0.489	0.402	0.345	0.3041	0.2736	0.2489	0.2293	0.2132	0.1995	0.1876	0.1685	0.1534
11.4	0.869−	0.772+	0.640	0.487+	0.400	0.344−	0.3028	0.2721	0.2475	0.228	0.2119	0.1983	0.1865	0.1675	0.1525
11.3	0.867+	0.771	0.638	0.486−	0.398+	0.342	0.3014	0.2707	0.2462	0.2268	0.2107	0.1972	0.1855	0.1667	0.1515
11.2	0.866	0.770−	0.636	0.484	0.397−	0.340+	0.2998	0.2692	0.2450	0.2256	0.2096	0.1961	0.1845	0.1656	0.1506
11.1	0.865	0.768	0.634	0.482	0.395	0.339−	0.2982	0.2678	0.2439	0.2245	0.2084	0.1949	0.1834	0.1645	0.1497
11.0	0.864	0.767	0.632	0.480	0.393	0.337	0.2966	0.2663	0.2428	0.2233	0.2072	0.1937	0.1824	0.1634	0.1488
10.9	0.863	0.766	0.630	0.478	0.391	0.335	0.295	0.265	0.2412	0.2218	0.2061	0.1926	0.1812	0.1624	0.1479
10.8	0.862+	0.7645	0.628	0.476	0.389	0.333	0.2935	0.2635	0.2395	0.221	0.205	0.1916	0.180	0.1614	0.147
10.7	0.862−	0.763	0.626	0.474−	0.387	0.3315	0.292−	0.262	0.238	0.219+	0.204−	0.1905	0.179	0.1605	0.1462
10.6	0.861	0.761	0.624	0.471	0.385+	0.330−	0.290	0.2605	0.237−	0.218−	0.2025	0.1895	0.178	0.1595	0.1455
10.5	0.860	0.759	0.622	0.469	0.384	0.328	0.288+	0.259	0.2355	0.217−	0.2012	0.188	0.177−	0.1585	0.1446
10.4	0.858+	0.757+	0.620	0.467	0.382	0.326	0.287	0.258−	0.234−	0.2155	0.200	0.187−	0.176−	0.1575	0.1435
10.3	0.857	0.756−	0.618	0.465	0.380	0.324+	0.285	0.256	0.233−	0.214	0.199−	0.186−	0.1745	0.1565	0.1427
10.2	0.856	0.754	0.616	0.463	0.378	0.323	0.2835	0.254	0.2317	0.213	0.1977	0.1846	0.1735	0.1555	0.1418
10.1	0.855	0.752	0.614	0.461	0.376	0.321	0.282	0.253−	0.230	0.2115	0.1965	0.1835	0.172+	0.1546	0.1407
10.0	0.853+	0.750	0.612	0.459	0.374	0.319	0.280+	0.251+	0.2285	0.210+	0.195	0.1825	0.171+	0.1537	0.1396

atmospheric pressure, in machines having more or less restricted inlet passages.

"Tables X and XI, provide data for quickly laying out in tenths of a pound the theoretical isothermal and adiabatic curves on indicator diagrams which start their compression anywhere between 14.7 and 10 lb. absolute. To prepare the indicator diagram for applying the tables, (see Fig. 9) draw horizontal pressure lines at 10-lb. intervals, to the scale of the indicator spring, using the portion *AP* of the diagram as a base line. Next, increase the length of the diagram by an amount equivalent to the percentage of the volumetric clearance in the cylinder at the end of the stroke, and erect the perpendicular line *BC*. Consider the length *AB* as one and divide it into 10 equal divisions. The tables give the horizontal measurements in percentages of one measured from the line *BC;* these locating the points of the compression curves on the various pressure lines.

"As an example, the sketch, Fig. 9, shows a normal indicator diagram from an air-cylinder compressing to 100 lb., the volumetric end clearance being 1 1/2 per cent., with compression starting at 1 lb. below atmosphere at sea-level; that is, at 13.7 lb. absolute. The diagram having been ruled with pressure lines and the subdivisions in length marked off, refer to isothermal values in Table X for 13.7 lb. absolute initial pressure. In the pressure columns will be found the horizontal measurements to be made on *AB* for the points in the compression curve. On the 20-lb. line this is 0.407, on the 40-lb. line 0.255, etc. The adiabatic values in Table XI for 13.7 lb. absolute initial pressure give, on the 30-lb. line 0.439, on the 50-lb. line 0.336, etc. Thus a sufficient number of points are located to readily and accurately construct the curves.

"The tables, being worked down to 10 lb. absolute pressure, may be used up to 10,000 ft. altitude, provided the inlet pressure does not start below 10 lb.

"The tables also show the approximate position (somewhere between the isothermal and the adiabatic curves) of the piston in percentages of its stroke, for any of the given pressures, and from the isothermal table may be seen the relative volume of air delivered at the given pressures as compared with the original volume, considered as 1, at initial pressure."

These adiabatic and isothermal curves when described are rather an aid to the eye in making comparisons with the actual compression-line of the indicator-card then necessary in computation. The mean effective pressure of the actual card is determined by its area, as ascertained by planimeter or otherwise, and the mean effective for adiabatic and isothermal compression under the same conditions may be found in Table IX, and the economy of the actual compression may be learned by comparison with them. This paragraph is only meant to apply to approximately sea-level computations.

TWO-STAGE COMPRESSION CARDS

Some one having asked in *Power* for "a diagram showing what constitutes a good card from a two-stage air-compressor" I offered a set of actual cards then recently taken from what I was disposed to call a good compressor, as compressors go.

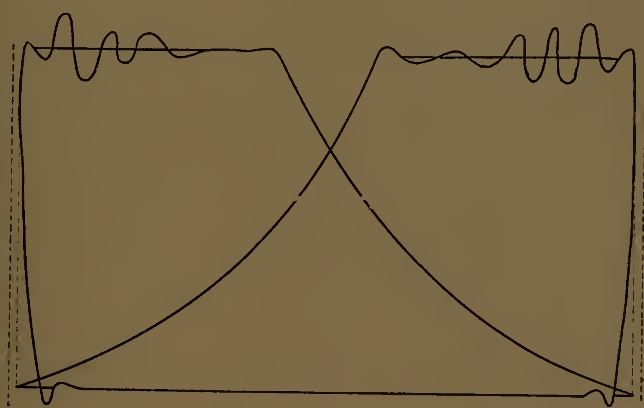

FIG. 10.—Cards from Low-pressure Cylinder.

These cards are here reproduced exact size, Figs. 10 and 11, the scale of the low-pressure cards being 15 and of the high pressure 60.

The diameters of the air-cylinders were respectively, 32 1/4 and 20 1/4 in., by 24 in. stroke; the diameters of the piston-rods of both cylinders were 3 3/8 in., and, both cylinders having the piston inlet, the outside diameters of the piston-inlet pipes were 10 3/8 in. and 6 7/16 in. respectively. The net piston areas for the low-pressure cylinder are:

Rear end (piston inlet), 732.32 sq. in.

Forward end (piston-rod), 807.92 sq. in.

The full cylinder capacity then per double stroke is:

$(732.32+807.92)\times 2$ (feet stroke) $\div 144=21.39$ cu. ft.,

or at 80 r.p.m., the actual running speed when the cards here shown were taken, 1711 cu. ft. per minute.

The cards from the low-pressure cylinder, Fig. 10, were traced as faithfully as possible, and in addition I have drawn a line in each approximately averaging the delivery line. The atmosphere line, which has disappeared in the engraving operation, was so close to the intake line as to be scarcely separable from it.

The indulations which so generally occur in the delivery-line of air-compressor cards are a great annoyance to say the least. They have been attributed to the use of free-moving poppet discharge valves, and on account of this various mechanically moved sliding or oscillating discharge valves have been devised, the real motive in the designing of which has too evidently been not the promotion of actual efficiency but the securing of a smoother looking card, and yet as ragged looking discharge-

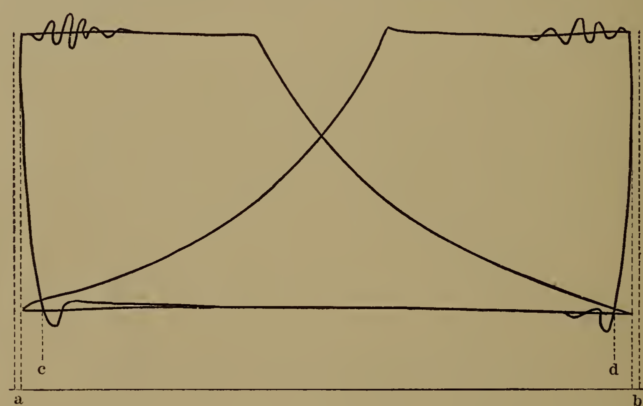

Fig. 11.—Cards from High-pressure Cylinder.

lines as I ever saw were on cards taken from a blowing engine with positively moved valves which gave a remarkably large and free passage for the air.

The objections to the mechanically or positively moved discharge valve are that the precise times for it to open must vary continually with the delivery pressure, and also to some extent with the speed of the machine, so that if the valve is timed to open for a certain pressure it will open too late for a lower pressure and too early for a higher pressure, causing in each case some loss of power by compressing above the delivery pressure in the one case, and by doing compression work over again when the air blows back in the other. The delivery pressure in the low-pressure cylinder of a two-stage compressor is, of course, constant, being determined by the relative capacities of the high-pressure and the low-pressure cylinders, but this does not relieve the mechanically moved discharge valve as to the more difficult proposition: to have it give free passage for the air right up to the end of the stroke, and yet to have it closed absolutely before the beginning of the return stroke, so that all the air compressed and expelled will be prevented from flowing back.

Of course it makes all the difference as to what the discharge valve is, its size and weight, its cushion, its spring and all that, which it will not do to go into in detail here. The merits of all the devices employed are fully set forth in the builders' catalogues. The promptness and efficiency of the poppet valves in closing in this case are sufficiently shown by the re-expansion-line, and the small clearance of which it gives sharp evidence.

In computing the volume of free air taken into the low-pressure cylinder and actually delivered, no deduction is called for in this case at the beginning of the compression stroke on account of deficiency of pressure, as the compression-line begins fully up to the atmosphere line. It is to be noted that this compressor not only has the piston inlet, but it has the hurricane piston-inlet valve, a recent and decided improvement upon the original piston-inlet valve. As the piston-inlet valve closes by its own momentum at the end of the stroke, it remains fully open until the actual end of the stroke—the hurricane valve opening being 12 per cent. of the piston area—and then closes instantly.

The amplitude of the opening provided by the hurricane valve is shown by the near coincidence of the intake line with the atmosphere line of the card. Although we know that the intake pressure in the cylinder during the stroke must be lower than the atmospheric pressure outside, to cause the air to flow in and fill the cylinder, the difference of pressure is here so slight that for most of the length of the stroke the intake line cannot be distinguished from the atmosphere line. At the precise moment of valve closure the momentum of the column of air which has been rushing through the inlet pipe is suddenly checked, but in the stopping of this rush there is a slight increase of pressure at the valve, which carries the pressure of the air in the cylinder fully up to or slightly above the atmospheric.

This phenomenon of the final inrush of free air at the close of the intake stroke, and the consequent accession of pressure at the moment when compression is to begin, has been observed in compressors of several makes and with different types of air admission, so that the fact seems to be indisputable if still incredible. The observed rise of pressure, however, has never been enough to be worth consideration except as a curiosity.

The actual free air delivered, as compared with the total capacity per stroke is as $a-b:a-d$. The total length $a-b$ of the card being 3.06 in. and the length of $a-d$ being 2.94, we have $3.06:2.94::1.00:0.9607$, or say 0.96, which may be called the actual volumetric capacity of the compressor. The free air actually delivered, then, per double stroke is

$$21.39 \times 0.96 = 2.53 \text{ cu. ft.}$$

It will be seen that the practice of deducting only the *actual clearance space* from the full cylinder content per stroke and making that the volumetric measure is not correct in the case of air compression, as the space in which this clearance air re-expands down to initial pressure is also to be included. The actual volumetric efficiency disclosed in this case is really quite high, although not quite as high as some *apparent* efficiencies reported.

We now look at Fig. 10, the cards from the high-pressure cylinder. The diameter of the cylinder being 20 1/4 in., the piston rod 2 3/8 in. and the piston-inlet pipe 6 7/16 in., the net piston areas are:

Rear end (piston inlet), 280.62 sq. in.

Forward end (piston-rod), 313.12 sq. ft.

The full cylinder capacity per double stroke, then, is

$$(280.62+313.12) \times 2 \text{ (feet stroke)} \div 144 = 8.24 \text{ cu. ft.}$$

Total length of card, 3.03 in.; total air-intake length, 2.93 in. Percentage:

$$2.93 \div 3.03 = 0.967.$$

Total intake:

$$8.24 \times 0.967 = 7.97 \text{ cu. ft.}$$

We found that the volume of free air taken into the low-pressure cylinder per double stroke was 20.53 cu. ft. This volume of free air at the intake pressure of the high-pressure cylinder, 23 lb. gage, would be (temperature constant)

$$23+14.5:14.5::20.53:7.93,$$

which is remarkably close to that of the high-pressure intake 7.97. The second cylinder at equal temperature should show slightly the less volume on account of probable slight leakages at stuffing-boxes and elsewhere. The actual result indicates that the intercooler brought the temperature almost, but not quite down to that at which the air entered the low-pressure cylinder. It will be noticed that I have assumed throughout 14.5 lb. as the atmospheric pressure, the compressor being located high enough above sea-level to warrant this.

CHAPTER VI

SINGLE-STAGE COMPRESSION

At the present writing—although no one may say how long it will hold true—most of the compressed air used comes from the reciprocating piston type of compressor, and this it will be proper for us to consider first. Where a few years ago it was quite common practice to use single-stage machines for working up to 80 or 90 lb. pressure the present more general limit is about 50 lb. What may be said here about single-stage compression will generally apply also to the first stage of compound compression.

Economy in air compression should begin at the beginning, and at the beginning we have to do with "free air," or air at atmospheric pressure. This is our raw material and in the keeping of accounts in the air-compressing business this raw material is generally measured and recorded not by weight but by bulk, so that first of all whatever quantity of air we use it is desirable to get it to the compressor in as small volume as possible. The smaller the relative volume of air at the beginning of the series of operations the greater will be the profit in the end for any service realized. The volume of free air increases or diminishes as its temperature rises or falls, which means that we should get our free air as cold as possible.

It seems necessary in all operations with compressed air to keep the accounts of profit and loss, and the record of work done, by the quantity of *free air* that is handled. This involves fewer uncertainties than if we were to base our computations upon the quantity of air after compression to any given pressure, or at any later stage in its transmission or use.

The individual air-compressor, when the necessary deductions have been made for clearance, for leakage, for valve action, for temperature proclivities, etc., when in fact its "personal equation" has been determined, makes a quite reliable air meter, and from it may be obtained a very close record of the free air taken in by it and compressed and delivered. After the

Fig. 12.—Rock Drills on Front of Capitol, Albany, N. Y.

beginning of operations the temperature of the air is such an uncertain and variable factor, and is still of such importance in the result, that all calculations may be upset by it. The absolute measure of the air operated upon would of course be its weight, but this it seems to be impossible to ascertain in extensive practical operations. While we may not deal with actual weight we may well bear in mind what will tend to increase or reduce the relative weight.

The volume of air at common temperatures varies directly as the absolute temperature. With our air-supply at 60° its absolute temperature is 521°, and the volume of it will increase or decrease 1/521 for each degree of rise or fall of temperature. In securing our supply of free air for the compressor, then, if we can get a difference in our favor of 5° by laying a pipe and leading the air in from the outside of the compressor-room, or from the shady side of the building, or from the coolest place near by, instead of using the air in the compressor-room, we accomplish a saving of about 1 per cent. If we secure a difference of temperature of 10°, which in practice is frequently quite possible, we save 2 per cent. absolutely without cost, except the first cost of the pipe or box to lead the air in. I know that the average machinist or engineer, or the man who calls himself distinctively the practical man, cannot commonly appreciate these small figures, or have any respect for such small savings, but when it comes to business I do not know why they should not have the same weight as the same values have in any other of the details of business. Brokers have to live and flourish upon commissions of 1/8 or 1/16 of 1 per cent.

The pipe to convey the cool, free air from the point where we determine to take it to the compressor may as well be of wood or of cement or earthenware as of iron, and in fact such material for its non-conductivity is to be preferred. The pipe should of course be large enough, and with easy curves instead of sharp angles, so as to convey the required flow of air with perfect freedom. Some of the best air-compressors of the day may be directly connected quite readily with an outside air-supply and they make provision for it; others cannot easily be so connected, which is unfortunate for them.

Filtering and Cooling Intake Air.—The air should not only be as cool as possible, but, for reasons in many cases more imperative than any power-saving conditions, it should be free from dust.

In perhaps the majority of modern and up-to-date installations some attempt has been made to have the air clean. The arrangement adopted has usually been merely to lead in an air conduit from outside the compressor room and to so protect the opening that cats could not be sucked in, or to pipe down from the roof with a wide-meshed hood over the top instead of taking the air from the room itself.

The means actually employed for most effectively filtering and cooling the intake air are generally improvised upon the spot and adapted to the special conditions in each case. An interesting and highly effective device for securing cool and pure

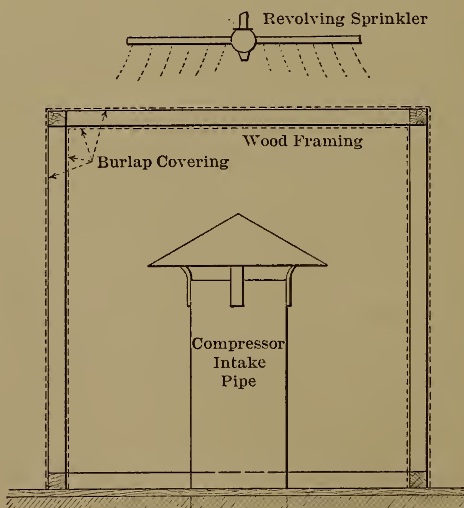

Fig. 13.—Air Strainer and Cooler.

air for compressors has been brought to my notice as in use at a plant of the Associated Oil Company, Kernville, California, in the Bakersfield oil district. This device or an equivalent was an absolute necessity there, as the region is hot and the air is loaded with dust. The device is eminently common-sensible and in use has proved so satisfactory that it is to be recommended for general adoption.

This air filter and cooler is located outside the compressor house, with the hooded intake air pipe standing up in the middle of it as shown in the sketch, Fig. 13. There is just a rectangular frame, or skeleton, made of light scantlings put together by a carpenter. The four sides and the top of the frame are covered with common burlap, tacked on, and the inside is similarly covered, with the thickness of the scantling between the two layers of burlap. This provides on every side two separate filtering surfaces for the air to pass through. Right over the center of the frame is located an ordinary revolving lawn sprinkler, with vertical axis, which slowly turns and keeps all the surfaces constantly saturated and dripping with water. The burlap in this case soon catches so much dust that it can be scraped off in considerable quantities, so that it is frequently necessary to souse the filter, inside and out, with a hose. The frame can be easily lifted off the intake pipe and turned over to expose the inside for the washing.

This device not only effectually separates the dust from the air but cools it, through the evaporation of the water, and the air enters the cylinder in unusually good condition. The intimate contact of the water with the air and the probable absorption of more or less water by the air has no effect upon the dryness of the air after compression. For in any case the compression and the subsequent cooling of the air will leave it more than saturated, so that there will surely be free water in the air to be gotten rid of, after the compression, and a little more or less will make no difference.

Another point which has not generally received the attention it deserves, although quite as important as the preceding, is the necessity, in the interest of the best power economy, of not only getting the air as cold as possible *to* the compressor, but of getting it as cold as possible *into* the compressor. We have too readily assumed that the one covers the other, when, as a matter of fact, it never does. The temperature of the air at the cylinder and about to enter it does not guarantee the temperature of the air in the cylinder at the moment when the cylinder is filled and compression begins. It is not too much to say that the temperature of the air outside the cylinder and of that inside is never the same. Yet it is not to be forgotten that the sole object of the effort to get cool air for the compressor is to have it as cool as possible, and of as small a volume as possible, at the moment when compression begins. How cool the air may have been at any previous moment, however near, has nothing to do with the case.

In another chapter I have remarked that the air-compressor is the ideal and the only perfect field for the use of the indicator, that it is the only place where the indicator diagrams will tell the whole story both of the power expended and of the work accomplished. This is undoubtedly true, but it is a statement that is quite likely to be understood to say more than it does say. The indicator diagram from the air-cylinder does not tell all that it seems to tell, or it sometimes tells it wrong. You may note upon the diagram the point, very near the beginning of the compression-stroke, where the cylinder, if we may believe the diagram, is filled with free air, or air at atmospheric pressure, and from that, after deducting what fills the clearance-space at the end of the stroke, we may compute the volume of free air actually compressed and delivered; and then, later, we may realize that we have not got the volume of free air that the diagram testifies to. This is due to the fact that the diagram has nothing to say about the actual temperature of the air, either at its admission, at its discharge, or at any point of the stroke.

With steam, unless it is superheated, the pressure indicated guarantees the temperature; with air the pressure and the temperature have no necessary connection. I might show you a diagram from an air-compressing cylinder where the air-admission line is almost exactly coincident with the atmosphere line, and where the compression line begins to rise above the atmosphere line immediately at the beginning of the compression-stroke, showing that the cylinder is completely filled with air at atmospheric pressure, and we may congratulate ourselves that the diagram is an excellent one in this respect; but suppose that when the cylinder is just filled, and compression is just beginning, our cylinder is filled with air at 120° instead of at 60°, which is the temperature of the supply. It means that our cylinder holds rather less than 0.9 of the air that we are assuming that it holds, and which the diagram says that it holds.

It means not merely that the practical capacity of the compressor is one-tenth less than we assume it to be, but that for the compression of this nine-tenths we are still expending the full power as represented by the steam-card. If the difference in indicated horse-power between the air-cylinder and the steam-cylinder is 10 per cent. of the air-cylinder, or if the power ratio of the steam to the air be 1.1:1, it is not a bad showing. But if this 1.1, the power of the steam-cylinder, is to be compared not with 1, the full capacity of the air-cylinder, but with .9, its actual contents, the case is quite different: 0.9:1.1::1: 1.22, which is a result not worth bragging about by any compressor builder.

There seems to be no means of ascertaining the actual temperature of the air during the operation of compression. The temperature of the air at different points of the stroke would be easily computable from the indicator diagram, which shows the pressure attained at any point, if we only knew the initial temperature, but as we have no means of knowing the initial temperature we do not know the actual temperature at any time. Who will tell us how to find it out? This does not seem to be an impossible problem. It looks at first sight almost as simple—not quite—as to tell how fast a stream of

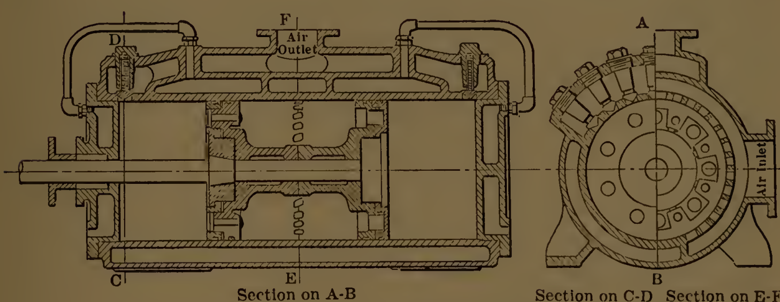

FIG. 14.—Novel Air Intake.

water flows through a pipe. But nobody has yet invented a perfect water-meter. In the meantime we can only use our mechanical judgment and common sense as to the best means of getting the air into the cylinder as cool as possible. We can say in a general way that the air should enter the cylinder by the shortest and most direct possible passage, and with as little contact as possible with any metal at a higher temperature than its own.

All this suggests and invites investigation and discussion as to the various means adopted by the different designers for getting the free air into the cylinder, the directness or tortuosity of the passages and the temperature of the surfaces which the air must pass, but this is not within the scope of the present publication. Some arrangements are undoubtedly better than others, and the survival of the fittest may be expected to work itself out here as elsewhere.

Fig. 14 is a sketch of the air-cylinder of a compressor of English manufacture which is presented here for whatever it may be worth. It appeals to the writer as likely to get the air into the cylinder somewhat cooler than is done by some other and more familiar arrangements, although some other features of the design may not be so admirable.

It will be seen that the cylinder is abnormally long for the piston stroke, and that the piston faces are widely separated, or in fact comprise two pistons fixed upon the same rod. A single air intake passage for both strokes surrounds the middle of the cylinder with numerous radial openings direct into the cylinder. The face of each piston carries a large annular valve of slight movement, covering liberal openings through the piston. This valve is quite heavy and is provided with a regular packing ring which fits the periphery of the cylinder, causing friction which operates the valves, thus opening or closing the piston passages upon each reversal of the stroke. This arrangment permits the most complete water-jacketing.

Returning now to the more familiar, or what we may call the standard type of single-cylinder compressor, and coming to the completion of the stroke when the air compressed has been expelled or delivered, the air remaining in the clearance-space between the piston and the cylinder head, both considerably heated by previous charges of air, although it may not have lost much of its heat of compression it must have lost some on account of the cooling influence of the water jacket, and its temperature cannot be quite as high as the theoretical temperature due to the compression. Still it is comparatively hot, and when it is remembered that this hot air becomes a part of the next cylinderful of air to be compressed it has been assumed that therefore the mean temperature of the contents of the cylinder is somewhat increased by this admixture. But this conclusion is hasty and unwarranted. This hot air in the clearance-space is only hot when under the terminal pressure, and as at this pressure it is not as hot as the theoretical temperature for the given compression it cannot upon its re-expansion to atmospheric pressure be as hot as it was before its previous compression began. It must be really somewhat cooler than the air that rushes in to fill the cylinder for the next stroke, and it therefore does not contribute any heat to the new charge of air, but rather receives some heat from it and thereby slightly cools it.

The face of the piston is exposed to the compression-heated air only until the re-expansion of the air in the clearance-space has occurred, and thus for almost the entire return stroke it is exposed to cooling rather than to heating influences.

The air remaining uncompressed in the clearance-space at the end of the compression-stroke, as it does not raise the temperature of the incoming air or tend to increase its volume, has therefore no bad effect in that respect, and in no way increases the power required for compressing a given quantity of air. The power that has been expended in the compression of this air in the clearance-space is not lost, or but a portion of it, as it gives out in its re-expansion, by helping the piston upon its return stroke, most of the power expended in its compression. Clearance in the air-cylinder, therefore, represents a loss of capacity in the air-compressor rather than a loss of power. And it is chiefly on account of its reducing the capacity of the compressor to compress its full quota of free air per stroke that it is desirable to keep the clearance as small as possible.

Leaky Discharge Valves.—In attempts to explain the causes of the ignitions and explosions which sometimes occur in air receivers and pipes, it is frequently suggested that the high air temperatures which are assumed to precede these occurrences are caused by leaky discharge valves, which allow a certain portion of the compressed and heated air to return into the cylinder retaining its high temperature there to be re-compressed and heated still more, this operation being repeated over and over until the air becomes so hot that if there is anything combustible or explosible in the neighborhood this red-hot air will be ready to touch it off. This "theory," variously worded, has been enunciated and repeated in technical journals of high standing with little protest from any source. No one has volunteered an explanation of how any portion of a cylinderful of air could be isolated from the rest and be churned back and forth in this way instead of mixing with and being carried along in its regular place in the procession, or how the air could continually re-expand as it leaked back and still retain its full high temperature.

Leaky discharge valves have still another scapegoat job. It seems to be a fact—at least indicator-cards say so—that in certain types of compressors—and there are more than one of them in which it occurs—if the intake air be led to the compressor through long unobstructed passages of not too great area, and if the inlet valves provide a free opening into the cylinder right up to the moment when compression begins, the cylinder may be filled with air quite up to and even a perceptible amount above atmospheric pressure, this high and untheoretical pressure being apparently caused by the sudden checking of the momentum of the rapidly moving column of air.

There are those who, while they cannot dispute the evidence of the indicator, refuse to accept the explanation. They say that any intake pressure quite up to or slightly above atmospheric pressure is not legitimately possible, and that when the indicator says that such pressure exists it can only be properly explained as caused by leaky discharge valves. The absurdity of this proffered explanation would seem to be self-evident.

The explanation is offered by respected professors and others in the presence of an entire indicator-card not otherwise abnormal. At the beginning of the return stroke the re-expansion-line drops promptly to slightly below atmosphere, showing that the discharge valves do not leak. Through the entire intake stroke the air line runs closely parallel to and slightly below the atmosphere line, showing that the discharge valves do not leak, but at the moment when there is the entire cylinderfull of air whose pressure theoretically should be slightly below atmosphere then there is such an inrush from the leaky discharge valves as to almost instantaneously raise the pressure of the entire cylinderful of air a half a pound or a full pound. When the compression stroke begins the compression-line is perfectly normal again, which it could not be if the discharge valves were still copiously leaking back. Obviously the sudden checking of the inrush of the air by way of the inlet pipe offers the more acceptable explanation.

Supposing that we are filling the air-cylinder by the natural inflow of the air under the pressure of the surrounding atmosphere, and that we have got into the cylinder the greatest possible actual weight or quantity of air under those conditions, and, assuming that the air is also as cool as we can get it, we may then be said to have got our material as cheaply as possible, to have started our business under the most favorable conditions, and with encouraging prospects; and we may then, and not until then, consistently and without reproach look for the available means of economy in the actual operation of compression. The same considerations that tend to economy in the procuring of the air, or of getting it into the cylinder, hold good also in all the subsequent operations of compression. The smaller the bulk or volume of any given quantity or weight of air the cheaper can the compression be effected and the better will be the economy; and, as the volume of the air at any given pressure depends upon its temperature, the supreme consideration throughout the operation is to keep the air as cool as possible. Keeping the air cool during compression means actually cooling the air during compression. No compression can be effected without a corresponding rise of temperature in the air compressed. Theoretically the rise will always be the same where the conditions are identical. Starting with a given volume of air and with the air at a given pressure and temperature, and compressing to another and higher pressure, the resulting volume and temperature should always be the same. Practically the temperature of the air after compression, or during compression, is never as high as the theoretical temperature, or as high as the books and tables say that it should be, and it is also widely variable under apparently slight changes of conditions.

This is not at all because the theory in the case is incorrect, but rather that it is incomplete, in that it is not cognizant of all the conditions that affect the case. Theory says, and correctly, that the element of time has nothing to do with the heat of compression; that a given volume of air when compressed to another given volume will have its temperature raised so much, whether it takes a minute, an hour, or a week to do it. Practically time has a great deal to do with the case. The readiness with which the air will receive heat from or impart it to whatever may be in contact with it, and the small amount of heat actually represented by its changes of temperature render the actual volume a highly elusive quantity, and time becomes a playground for it.

In a compressing-cylinder in actual use all the parts of it, the body of the cylinder, the heads, the piston and rod, the valves and seats or guides become heated by their contact with the compressed air; but while they are thus becoming heated they are only heated by this contact, and while being heated they are also being cooled, as they are constantly transmitting some of the heat received from the air and dispersing it by conduction or radiation; and, consequently, these parts are never as hot as the air that heats them—when the air is at its hottest—and the air also is not as hot as it would have been but for its contact with them. The metallic parts after a time of continuous operation attain an average temperature, and will not get any hotter. The mean temperature attained will depend upon the facilities provided for taking the heat away. Nothing better is known or has been suggested for conveying away the heat than cold water.

When the entire compression is effected in a single cylinder the heat of compression is abstracted from the air mostly at the latter part of each stroke, when the air is at its hottest and when the difference in temperature between the air and its surroundings

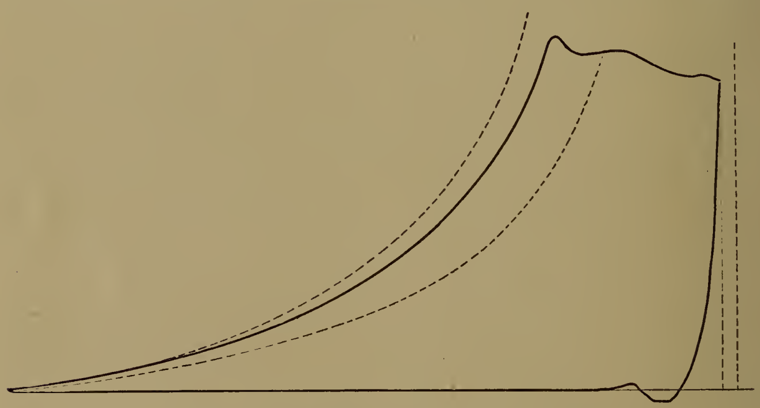

Fig. 15.—When Compression Line Leaves the Adiabatic.

is the greatest. Indeed it is to be supposed that in active compression the air loses none of its heat of compression during the earlier part of the stroke unless the means of cooling the cylinder parts are unusually efficient and operative. If at the beginning of the stroke the cylinder is hotter than the air, as it naturally must be, the air is naturally heated rather than cooled by the contact. Practical evidence of this is not wanting. Indicator diagrams from air-compressing cylinders are easily to be found, as Fig. 15, where the compression-line of the diagram does not leave the adiabatic line until the first quarter of the stroke is traversed. In this connection it may be remarked that for evidence upon the point that we are considering any indicator-cards that are taken when a compressor has just been started, and before the cylinder parts have attained their full average temperature, are not to be considered. Such cards promise better than the actual performance of the compressor will fulfil.

The heating of the air does not continue throughout the whole stroke of the piston, but is accomplished and ceases at the moment that the full pressure is reached; and for the remainder of the stroke, while the compressed air is being ejected from the cylinder, the air is becoming somewhat cooler, while the metal inclosing it is becoming hotter. The heat of the cylinder parts is not evenly distributed. The ends of the cylinder and the entire cylinder-heads, being exposed to the air when it is hottest, naturally become hotter than the middle of the cylinder, which never feels the hottest air. The importance of the water-jacket, in the absence of any better cooling device, is obvious enough. The cooling effect of the water is greater when it is applied to the cylinder-heads than anywhere else, because they are exposed to the heated air for the greater portion of the stroke, while the inner surface of the cylinder itself is covered by the advancing piston. Apart from the cooling of the air under compression, and the reduction of its volume, the water-jacket is a necessity as affecting the lubrication of the cylinder surfaces. Without some such means of cooling the cylinder it would become so heated as to burn the oil and render it useless as a lubricant.

The device of cooling the air by the injection of a spray of water into the cylinder is probably the most effective cooling arrangement that has ever been devised, but collateral objections have driven it completely out of use in all new compressors at least, in the United States. When the spray is used the success of it as an air-cooling agent is entirely dependent upon the mode of its application. The spray can only possibly effect the intended purpose when diffused through the air while it is being compressed, or during the compression-stroke of the piston. It can only cool the air while it is hot, or while it is being heated; so that to admit the water with the incoming air is only to let it fall inert and useless to the bottom of the cylinder, to be driven out by the piston. Air so admitted may have a *quasi* usefulness in filling the clearance-space at the end of the stroke, but it can do little or nothing toward cooling the air. The presence of the water may also make it unsafe to run the compressor at a speed that would be otherwise safe and proper. With the use of water in the compression cylinder, whether properly injected or not, no satisfactory means of lubricating the surface of the cylinder has ever been found, so that the friction of the piston and the loss of power by that means is greater than with other systems of compression. The piston and cylinder surfaces also wear away rapidly, so that the repair costs and the inconveniences entailed are greater than with other systems.

Other things being equal, a cylinder of small diameter has a decided advantage over a large one in cooling the air during compression. In a large cylinder the portion of air immediately

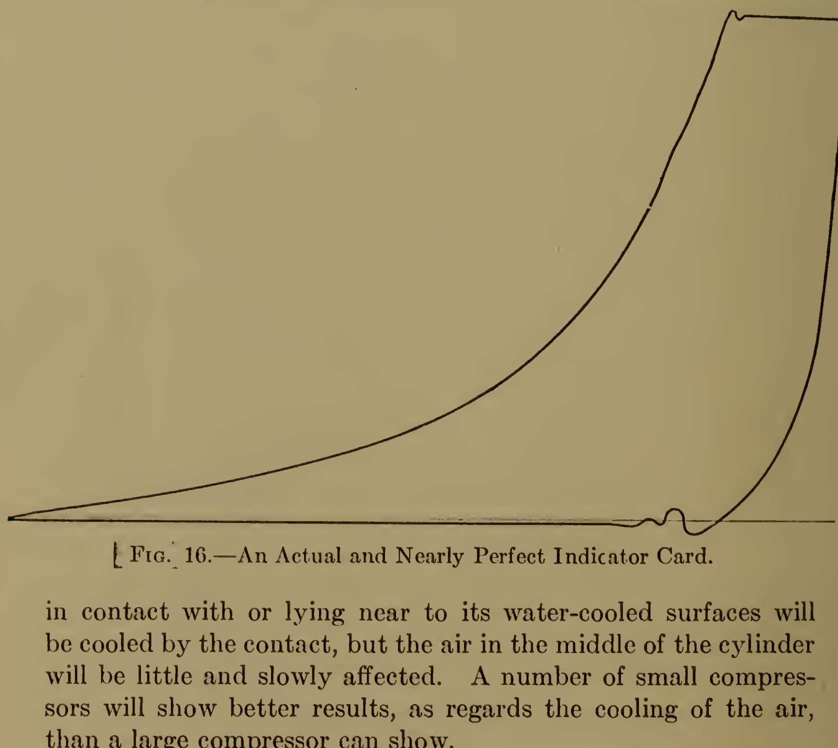

Fig. 16.—An Actual and Nearly Perfect Indicator Card.

in contact with or lying near to its water-cooled surfaces will be cooled by the contact, but the air in the middle of the cylinder will be little and slowly affected. A number of small compressors will show better results, as regards the cooling of the air, than a large compressor can show.

The indicator-card here reproduced, Fig. 16, which came into my possession more than a score of years ago, is still the best and most satisfactory diagram from a single-stage compressor which I have ever seen. It was taken from one of a series of small compression-cylinders entirely submerged in a tank of water. The scale of the diagram is 30.

CHAPTER VII

TWO-STAGE AIR COMPRESSION

In the preceding chapter only single-stage compression was considered. When air is to be compressed to pressures above 40 or 50 lb. and up to 200 lb. or so, two-stage compression should be employed, and within the range suggested the higher the pressure the greater the need of it. The two-stage plan is adopted for the sake of the cooling of the air and the consequent reduction of volume before the second compression, so the inter-cooling is the essential detail of the operation, and without that two-stage compression could be of no advantage.

Not only is there generally an appreciable saving of power in two-stage compression, but, perhaps more important, is the avoidance of the high temperatures, thus permitting more satisfactory lubrication, greatly reducing the deposition of combustible material in air receiver and pipes and minimizing the liability of fires and explosions. Single-stage compression seems to be responsible for most of the explosions which occur, as will appear elsewhere.

The first point to be considered in two-stage compression is the equitable distribution of the load between the two cylinders, which would be determined by their relative capacities, and these required relative capacities would vary with the ratio of the initial and terminal air-pressures. If the pistons of both the air-cylinders have the same stroke then their volumetric capacities will be as the squares of the cylinder diameters, and the ratios of these should be as the square root of the number of compressions required.

Thus in working at sea-level and compressing and delivering air at 90 lb., the ratio of pressures would be $(90+14.7) \div 14.7 = 6.44$ and $\sqrt{6.44} = 2.53$. Then say that our intake or low-pressure cylinder was 20 in. in diameter, to find the diameter of the high-pressure cylinder: $20^2 = 400$, $400 \div 2.53 = 158$ and $\sqrt{158} = 12.56$ for the diameter of the high pressure, or say 12 1/2 in.

The concise formula for the abve operation is:

$$d=\sqrt{\frac{D^2}{\sqrt{\frac{P_1+P_2}{P_1}}}}$$

Here D is the diameter of the low-pressure cylinder
d is the diameter of the high-pressure cylinder
P_1 is the initial absolute air-pressure
P_2 in the terminal absolute air-pressure

It is evident that the ratios of cylinder capacities must vary with the fixed working conditions, and a compressor properly adapted for sea-level work and for comparatively low delivery pressure would be far from its best if working at a high altitude and high delivery pressure.

Computing by the above formula, if we had a low-pressure cylinder 30 in. in diameter working at sea-level to compress to 80 lb. the diameter of the high-pressure cylinder would be 18.84 in.; while to compress to 120 lb. the diameter would be 17.26 in. Working at an altitude of 10,000 ft. (normal air-pressure 10 lb.) and compressing to 120 lb. the high-pressure cylinder diameter would be 15.81 in. In building compressors it is not customary to work to any computed cylinder diameters nearer than 1/4 in., and quite generally the nearest whole inch is the figure adopted.

Three-stage Compression. Ratios.—For three-stage compression, which is advisable for pressures up to 1000 lb., the cube root instead of the square root of the ratio of initial and terminal air-pressures becomes the ratio of the successive cylinder capacities, the formula then being:

$$d=\sqrt{\frac{D^2}{\sqrt[3]{\frac{P_1+P_2}{P_1}}}}$$

Here D is the diameter of the first cylinder when computing that of the second, but it is also that of the second cylinder when computing the diameter of the third, and d is the diameter of either the second or the third cylinder, as computed from that of the cylinder which precedes it.

Say that we have a cylinder 30 in. in diameter as before, this diameter being fixed upon for the free-air capacity of the machine, to compress to 1000 lb. the ratio of initial and terminal absolute pressures will be 70, and $\sqrt[3]{70}=4.12$ which will be the ratio of cylinder areas. Then $30^2 \div 4.12=218$ and $\sqrt{218}=14.76$ diameter of second cylinder. The area of this cylinder being 218, then we have $218 \div 412=53$, and $\sqrt{53}=7.28$, diameter of third cylinder.

Four-stage Compression. Ratios.—For four-stage compression the fourth root of the ratio of initial and terminal absolute pressure is taken as the ratio of the successive cylinder capacities, the formula then becoming:

$$d=\sqrt{\frac{D^2}{\sqrt[4]{\frac{P_1+P_2}{P_1}}}}$$

Here as before D is the diameter of the largest cylinder and also of the two following cylinders as compared with the next smaller one and d is the diameter of each smaller cylinder as compared with its immediate predecessor.

With an intake cylinder 20 in. in diameter what will be the successive diameters of the other cylinders to compress by four stages to 2500 lb.?

$2514.5 \div 14.5=173$, the ratio of absolute pressures.

$\sqrt[4]{173}=3.62$ ratio of successive cylinder capacities.

$400 \div 3.62=110$ $\sqrt{110}=10.48$ diameter of second cylinder.

$110 \div 3.62=30.4$ $\sqrt{30.4}=5.5$ diameter of third cylinder.

$30.4 \div 3.62=8.4$ $\sqrt{8.4}=2.9$ diameter of fourth cylinder.

If, as often is the case, the last cylinder was single acting, then its diameter would be $\sqrt{8.4 \times 2}=4.1$

The Distribution of the Load.—In stage compression the proper determining of the relative cylinder capacities is only a part of the problem; the location or arrangement of the cylinders to secure an equitable distribution of the load is also of importance.

It happened that nearly at the beginning of my apprenticeship to compressed air I was brought into rather intimate contact with a two-stage compressor from which many things were to be learned—some of them to be unlearned later.

This was a so-called straight-line machine with single-acting tandem air-cylinders, the two air pistons upon the same rod with the steam piston, and the alternate work of the two air-cylinders being done upon the alternate strokes of the steam-engine. The air-cylinders were 20 in. and 11 3/4 in. respectively and the stroke 18 in. The capacity ratio of the two cylinders, deducting the area of the piston rod in the larger cylinder was 1:0.35.

Indicator-cards were taken from both air-cylinders with the compressor delivering air at 35 lb., at 40 lb., and then by intervals of 10 lb. all the way up to 120 lb. The cards here presented

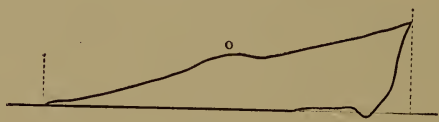

Fig. 17.

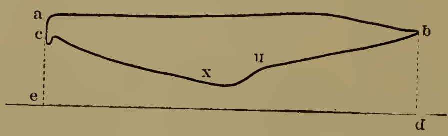

Fig. 18.

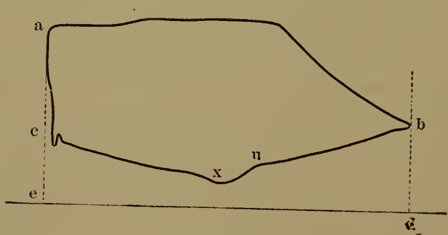

Fig. 19.

Figs. 17–19.—Indicator Cards from Two-Stage Single Acting Tandem Cylinders.

are as good as a greater number for bringing out the peculiarities of the case. Fig. 14 is from the first or low-pressure cylinder. This card did not vary in any particular throughout the whole series from 35 lb. to 120 lb., and it would have continued the same no matter how high the terminal or delivery pressure of the second cylinder were carried. A tracing was made of one of these cards and laid over several others of the series, and the variation was so slight as to be scarcely discoverable at any point.

The mean effective pressure of Fig. 17 is 15.8 lb., and the terminal pressure is 35 lb. While the terminal pressure in this first cylinder is 35 lb., it does not mean that if the two-stage compressor were compressing and delivering air at 35 lb. gage, the first cylinder would be doing all the work of the compressor. It is to be remembered that the complete work of air compression comprises two distinct operations: the compression of the air to the required pressure, and the expulsion or delivery of the air against practically the same pressure in the air-pipes or in the air-receiver. In the case that we are now considering, where the air is delivered from the compressor at a pressure of 35 lb., the first cylinder happens to do all of the work of compression, and none of the work of expulsion or delivery. In any case of two-stage compression, if either cylinder is to be called distinctly the "compressing" cylinder, that term always belongs to the first cylinder rather than to the second. If our two-stage compressor were delivering air at a pressure higher than 35 lb., the first cylinder would still compress the air to 35 lb. as before, or would do only a portion of the total compression, and of course none of the delivery. The height to which the first cylinder will continually compress the air is determined by the relative capacities of the two cylinders, modified to some extent by the cooling of the air that may be effected in its passage from one cylinder to the other. The work of the second cylinder when the compressor is delivering the air at 35 lb. is shown by Fig.18, taken from that cylinder. The delivery-line *ba* in this case would be a perfectly horizontal line if the movement of the piston were uniform throughout the stroke, the rise and fall of the line corresponding approximately to the acceleration and retardation of the piston.

At whatever pressure the compressed air may be delivered by the compressor the mean effective pressures for the two distinct operations of compression are never alike. The mean effective pressure for compression only is always lower than the M.E.P. for delivery only, and of course also lower than for the combined operation of compression and delivery as performed in a single cylinder. In the compression Table IX, columns 6 and 7 give the mean effective pressures for the whole stroke when all of the work of compression and delivery is done in a single cylinder, column 6 being for isothermal and column 7 being for adiabatic compression. In the same table columns 8 and 9 give respectively the isothermal and the adiabatic M.E.P. for the compression part only of the stroke of a single air-cylinder.

Resuming now our compound compression, and referring again to Fig. 17, we notice that its mean effective pressure—15.8—is greater than the pressures given in either columns 8 or 9 for compression only to 35 lb., where the entire work of the compressor is done in a single air-cylinder. The table referred to, as we have previously stated, has nothing to do with compound compression, but the comparison of figures might provoke suspicion that in compound or two-stage compression we are doing the same work of compression as in the single air-cylinder, but at greater expense, and it is therefore proper to refer to it here.

The case represented is different in more than one particular. In single-stage compression the compression is all done in the one cylinder, and throughout the entire compression-stroke the same quantity of air is acted upon. In Fig. 17 we are not doing the entire compression part of the work in the one cylinder, although it is begun there, and the weight of air acted upon is not the same throughout the stroke. While at the beginning of the stroke the air acted upon is the free air contained in the first cylinder and just admitted from the atmosphere, this continues only for the first half of the stroke, and for the latter part of the stroke the whole body of air then undergoing compression consists not only of all the contents of the first cylinder that have not been expelled by the advancing piston, but also of the entire contents of the passage connecting the two cylinders, and the contents of that part of the second cylinder which has been vacated by its retreating piston. Fig. 17 shows the compression beginning at *a*, at the beginning of the stroke, and with the free air contents of the first cylinder alone.

This goes on until the point *o* is reached, near the middle of the stroke, and then communication is opened with the air-passage that connects the cylinders, and through that with the second cylinder. When the previous compression-stroke of the first cylinder ended, the passage connecting the cylinders was filled with air compressed to 35 lb., and by the action of the valves this passage was then for a time shut off from communication with either cylinder. This passage, in fact, remains shut off from communication with either cylinder during the whole of the return stroke, while the first cylinder is being filled with a fresh charge of free air, and while the compressed air in the smaller cylinder is being expelled into the discharge-pipe and the air-receiver. When the return or intake stroke of the larger cylinder has ended, which return stroke is the delivery-stroke of the smaller cylinder, and when the compressed air has all been expelled from the smaller cylinder by its piston reaching the end of it, then the return stroke of the smaller cylinder commences, this stroke being of course coincident with the next compression-stroke of the larger cylinder.

With the commencement of the return stroke of the smaller piston the air confined in the connecting passage begins to re-expand and to flow into the smaller cylinder. The pressure is thus falling in the air-passage, on account of its supplying the smaller cylinder, and at the same time compression is going on in the larger cylinder, and the pressure in it is rising. These simultaneous operations go on until at length the point *o* is reached, where the pressure in the larger cylinder exceeds the pressure in the air-passage and in the smaller cylinder, and the air from the larger cylinder begins to flow into the air-passage, and at the same time the entire contents of the air-passage and of the smaller cylinder become constituent parts of the body of air that is being compressed by the advancing piston of the larger cylinder, and thereafter until the end of the stroke the compression of the combined contents of large cylinder, air-passage, and small cylinder goes on together. The last one-third of the compression-stroke in Fig. 17 and the portion *ub* in Figs. 18 or 19 represent the same operation of compression, the line in Fig. 17 showing a somewhat higher pressure than in Fig. 18 or 19 on account of the friction to be overcome in passing the valves and passages.

The mean effective pressure for the combined operation of compressing and expelling the air at 35 lb., or for the whole operation of air compression so termed, when performed adiabatically in a single cylinder is, theoretically, 21.6 lb. Practically, without any special arrangements for cooling the air, the M.E.P. usually falls somewhat below the above figure, as the air inevitably loses more or less of its heat during the operation. If we consider Fig. 17 in connection with Fig. 18, they together represent the whole operation of compression to 35 lb. by two-stage compression. Fig. 17 representing the compression of the air and Fig. 18 representing its expulsion or delivery. The mean effective pressure of Fig. 17 is, as we have seen, 15.8, and that of Fig. 18 is 16.4 lb. But it must be remembered that the diameters of the two cylinders are quite different, and 16.4 lbs. in the 11 3/4 in. cylinder is only equal in power to 5.65 lb. in the 20 in. cylinder, and 15.8+5.65=21.45 lb., a mean effective pressure quite close to what might have been expected for the entire operation of compressing air to 35 lb. without any device for cooling the air. When we remember that the use of two cylinders instead of one for the same operation of compression means necessarily a greater first cost for the apparatus, to the builder if not to the purchaser, a larger number of parts, increasing the liability to accidents and delays, and a greater amount of friction, both in the air and in the machine, to be constantly overcome, it is evident that two-stage compression of itself costs more than single-stage compression.

While these diagrams were being taken the compressor was run at about 80 r.p.m., or 240 ft. of piston travel per minute, throughout. At this speed the indicated horse-power of Fig. 17 for the first cylinder is 18.05 and that of Fig. 18 from the second cylinder is 6.46, their sum being 24.51. Fig. 19 is from the smaller cylinder when compressing to 70 lb. The M.E.P. of Fig. 19 being 43.4, and the indicated horse-power being 17.1, the i.h.p. for Fig. 17 being, as before, 18.05, their sum is 35.15. When compressing and delivering air at 70 lb., as indicated by Figs. 17 and 19, it will be noticed that the i.h.p. of the two cylinders is nearly equal, and it would thus seem that the ratio of the cylinder capacities to each other was approximately correct for that pressure. The relative diameters and areas of the two cylinders may have been determined upon this assumption, which involved an ignoring of essential particulars.

The arrangement of the tandem, single-acting, two-stage compressing cylinders is about as bad a one as could be devised for an air-compressor, and no possible change in the relative capacities of the two cylinders can make it right. The trouble in the case is that while the sum of the indicated horse-powers as computed from the actual enclosed areas of the two cards is correct as representing the total horse-power consumed in the operation, it does not correctly represent the actual distribution of the resistances as encountered in the opposite strokes of the engine. The back pressure in the second cylinder, which thus far has not been thought of, imperatively demands recognition and accounting with, as modifying the total resistances encountered. The back-pressure line, or, perhaps more correctly, the return-pressure line, *cxub*, as we have seen, starting at *c*, represents for nearly one-half the stroke the re-expansion of the contents of the air-passage. This re-expansion goes on in the passage and in the smaller cylinder combined until the point x is reached, when the compression going on in the larger cylinder has brought its contents up to the same pressure. Then after a short interval, *xu*, occupied in securing a sufficient excess of pressure, and in reversing the movement from expansion to compression, the compression continues from u to the end of the stroke, when the pressure of 35 lb. is again reached.

As the whole of Fig. 17 is always the same, no matter what may be the working pressure of the compressor, so that it is not below 35 lb., so also the return line of the diagram from the second cylinder is always the same, and the only change in the pair of Figs. 17 and 18 or of 17 and 19 for different delivery-pressures is in the upper line *ba*, the compression- and delivery-line of the second cylinder. When compressing to 35 lb. only, there is no compression in the second cylinder, and its whole stroke is occupied in delivery. At the beginning of the stroke the resistance against the high-pressure piston is represented by the height of the vertical line *bd*. The resistance at any point of the stroke would be represented by a vertical line at that point drawn from the line *ba* down to the atmosphere-line, and the total resistance for the working-stroke is represented by the enclosed area, *bdea*. This means that the total back pressure, *bdec*, is to be added to, or, rather, is not to be deducted from, the work of the compression- and delivery-stroke of the high-pressure cylinder. During this working-stroke of the high-pressure cylinder the low-pressure piston is making its return stroke and allowing its cylinder to refill with air at atmospheric pressure.

The pressure upon each side of the low-pressure piston upon its return stroke is practically that of the atmosphere, and therefore no resistance of any magnitude is to be taken into account as increasing or diminishing the total work of the high-pressure cylinder for its delivery-stroke. When, however, the low-pressure cylinder is doing its work of compression, it is assisted in its work by the return or back pressure of the high-pressure cylinder, which acts upon the high-pressure piston in the same lineai direction as the low-pressure piston is travelling. The back pressure, *bdec*, which is added to the work of the high-pressure cylinder for its delivery-stroke, as represented by the enclosed area *bac*, is to be deducted from the work of the low-pressure cylinder for its compression-stroke as represented by Fig. 17.

If now we go over the series of indicator-cards, computing the indicated horse-power of each, adding the i.h.p. of the back pressure to the i.h.p. of each of the high-pressure cards, and deducting the same from the i.h.p. of the low-pressure card, as above described, we find that the net resistance for the alternate strokes is very inequitably distributed. The figures for compressing to 120 lb. are also given to aid the comparison although the delivery or high-pressure card for that pressure is not shown. The case will stand like this:

M.E.P. of low-pressure cylinder 15.8 lb., i.h.p. 18.05.

M.E.P. of return stroke of high-pressure cylinder 20.1, i.h.p. 7.88.

Then $18.05-7.88=10.17$, the constant net i.h.p. for the compression-stroke of the low-pressure cylinder or the return stroke of the high-pressure cylinder.

M.E.P. of high-pressure cyl. at 35 lb. 16.4, i.h.p. 6.46.

M.E.P. of high-pressure cyl. at 70 lb. 43.4 i.h.p. 17.1

M.E.P. of high-pressure cyl. at 120 lb. 65.7 i.h.p. 25.89.

Then adding to these results the i.h.p. for the return stroke, which should not have been deducted from the delivery-stroke, we have:

$6.46+7.88=14.34$ when delivering at 35 lb.
$17.1+7.88=24.98$ when delivering at 70 lb.
$25.89+7.88=33.77$ when delivering at 120 lb.

As these several results for the delivery-stroke are successively to be compared with the constant i.h.p. 10.17 for the initial compression-stroke, it will be seen that even when delivering the air at but 35 lb. the delivery-stroke of the high-pressure cylinder takes nearly 1 1/2 times the power required for the return stroke. When compressing to 70 lb. under the above arrangement the delivery-stroke takes nearly 2 1/2 times the power of the return stroke, and when compressing to 120 lb. it takes more than 3 times as much.

The total power required for the above compressor at the speed given is:

35 lb. $10.17+14.34=24.51$
70 lb. $10.17+24.98=35.15$
120 lb. $10.17+33.77=43.94$

The volume of free air compressed and delivered at either pressure is 262 cu. ft. per minute.

The loss by friction in a two-stage compressor should be greater than in a single-stage compressor of the same free air capacity and working to the same pressure, and the total friction of single-acting cylinders must be proportionately greater than that of double-acting cylinders, so that if for a common single-stage double-acting compressor we allow 10 per cent. for the total friction of the machine, it is probable that 15 per cent. is not too great to allow for the arrangement that we have been considering above.

CHAPTER VIII

TWO-STAGE AND THREE-STAGE COMPRESSION

There may be those who could think that in the preceding chapter most of the space had been wasted, inasmuch as the matter presented was more in the nature of an explanation of "how not to do it," as any case of single-acting, two-stage tandem compression must be. Having gone so far, however, it may be proper to go a little further to satisfy ourselves as to the advantages or otherwise in the double-acting tandem arrangement.

Having the single-acting, two-stage tandem arrangement still in view, it is to be noted that when the compressor is in operation both pistons are always exposed to atmospheric pressure upon the sides nearest to each other. The other side, or the compressing side, of the larger piston is also exposed to atmospheric pressure, or very near it, during its intake stroke. The compressing side of the smaller piston is never exposed to atmospheric pressure when the compressor is in operation. During the intake stroke of the smaller cylinder, while it is receiving the air which is being compressed in the larger cylinder, its piston is subject to the pressure that is due to that initial compression. As both of the pistons are upon one rod, whatever pressure there may be against the smaller piston when the larger piston is doing its work is just so much help for the larger piston, and consequently *cbde* of Fig. 19 is to be deducted from the total work of Fig. 17.

In Fig. 20 the area *cbde*, representing this reacting pressure, has been reduced to the scale corresponding to the relative area of the larger cylinder, and has been superimposed upon Fig. 17. It will be seen that until the point *i* is reached the steam cylinder, or whatever motor is employed, has "less than nothing" to do, and if the compressor were running slowly, it would be apt to give a perceptible jump ahead just after passing this center. This has been actually observed to occur in a compressor of this type. In Fig. 21 the two diagrams have been combined into a single figure, with AB as the line of no resistance. This, it will be remembered, represents the distribution of the resistance for the compression-stroke of the larger piston. For nearly one-quarter of the stroke, considering here the air-cylinders only, and with no reference to the driving power of the steam-cylinders, the larger piston has a force behind it greater than the resistance

FIG. 20.

FIG. 21.

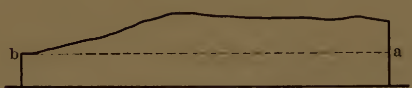

FIG. 22.

FIG. 23.

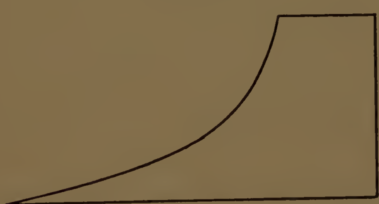

FIG. 24.
Distribution of the Load.

in front of it. From the point i the net resistance begins to rise before the larger piston, and continues to rise until the extreme end of the stroke, except for a slight interval at the middle. Fig. 22 represents the resistance for the return stroke, which is the delivery-stroke of the smaller piston. This diagram is the same as *baed* of Fig. 18, but drawn to the scale of the larger cylinder for comparison. It has also, for convenience of comparison by the eye, been reversed endwise.

It is easy enough by a glance at Figs. 21 and 22 to see the difference in the resistances for the alternate strokes. If the compressor were delivering the air at 35 lb., instead of at 70 lb., the upper line of Fig. 22 would approximately follow the dotted line *ba*, and the resistance would be practically uniform for the entire stroke. Fig. 21, representing the alternate stroke, would remain precisely the same whether the smaller cylinder were delivering the air at 35 lb., at 70 lb., or at any higher pressure, and even at the lower pressure the resistance for this stroke would not be as great as for the delivery-stroke.

It is evident that the resistance for the alternate strokes could not be equalized by changing the relative capacities of the two cylinders. To reduce the smaller cylinder would indeed tend toward an equalization of the resistances by allowing the first cylinder to do more work and compress the air to a higher pressure; but to raise the pressure in the first cylinder would be to defeat the purpose for which the two-stage compression is adapted, that of allowing a cooling of the air and a reduction of its volume before its compression is too far advanced.

As Figs. 21 and 22 represent the resistances for the alternate strokes of single-acting cylinders, these resistances may be added together and we may combine them, as is done in Fig. 23, and we then have the diagram for either stroke of tandem double-acting cylinders of the same sizes. This of course represents double the free air capacity of the single-acting cylinders. Fig. 24 is a theoretical diagram of a double-acting single-stage compression cylinder of the same capacity, the assumed compression-line being the mean of the adiabatic and the isothermal curves.

The maximum resistance for the stroke in the two-stage double-acting compressor is only three-fourths of the maximum resistance for the single-stage compressor. The resistance at the beginning of the stroke is not as low in the former as in the latter, and the distribution of the resistance over the whole stroke is decidedly more uniform. As to the total effective resistance for the stroke, as we have here developed it, the two-stage, compressor shows no advantage over the single-stage even while ignoring the additional friction of the former. In fact, the mean effective resistance of Fig. 24 is somewhat less than that of Fig. 23. This might have been expected, because in the cylinders from which Fig. 23 was evolved the full benefits of water-jacketing were not employed, the cylinder heads, for instance, not being jacketed at all.

Two-stage Cross Compound.—The indicator-cards, Figs. 25 and 26, though in several particulars not ideal, still speak for themselves as to the equitable distribution of load which is

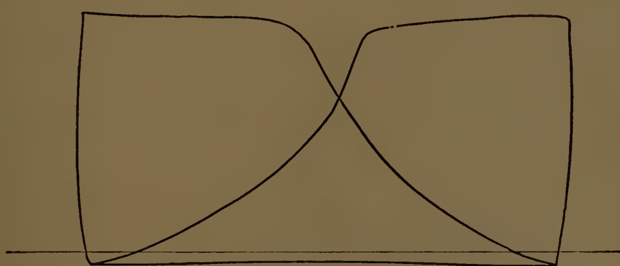

Fig. 25.—Low Pressure, Two-stage Double Acting Cross Compound Compressor.

possible in two-stage compression, without inviting any questioning such as the machine with the single cylinders provoked.

In this case each double-acting air cylinder is tandem to a steam cylinder of a cross-compound machine, the cranks of which are at right angles, the air between the stages passing

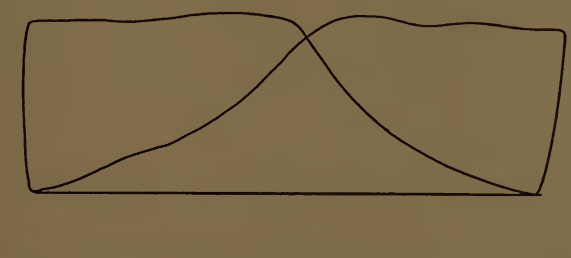

Fig. 26.—High Pressure, Two-stage, Double Acting Cross Compound Compressor.

across from the low-pressure to the high-pressure cylinder through an efficient intercooler which with its connections has a large air capacity as compared with the cylinder volume.

The low-pressure air-cylinder was 31 in. in diameter and the high-pressure cylinder 19.5 in. in diameter with a stroke of 42 in. and at the leisurely speed of 40 revolutions per minute a piston speed of 280 ft., which is not more than one-half the speed at which many compressors are run to-day. The scale of the first card is 20 and that of the second card is 60.

These cards taken together represent a remarkably even distribution of the load throughout the entire revolution. A little over one-half of each stroke of each cylinder is consumed in compressing the contents while the other half of the stroke is occupied in expelling the charge at the full working pressure for that cylinder. The compression portion of the stroke of one cylinder is coincident with the delivery part of the stroke of the other cylinder, and these occur together for each quarter of the revolution, so that the fluctuations of torque are much less on account of the air-cylinders than for that of the steam-cylinders, and the air cylinders thus help to promote rather than to defeat steady running.

The low-pressure cylinder delivers air to the intercooler for less than one-half of each stroke, while one or the other of the ends of the high-pressure cylinder is taking air nearly all the time, but the fluctuations of intercooler pressure are almost imperceptible, and all the air passing through is at practically the highest inter-cylinder pressure, or in the best condition for cooling.

A Study of Three-stage Compression.—Having the formulas provided, with the detailed requirements as to volume of air, ultimate pressure, etc., it would seem to be a simple thing to compute the cylinder dimensions and other particulars and then to proceed to design and build the machine. Before we get very far, however, we find that there are many details to be decided upon, possible different arrangements to be selected from, with no absolute best in sight.

The compressing of air is not in any case as simple an operation as the pumping of water, and when high pressures are required, involving, as in this case, multiple-stage compression, the problem of equitably apportioning the work and the effect for each step of the operation, the providing for the easy flow and the efficient cooling of the air between the stages, the reduction of machine friction to a minimum, the providing for the proper lubricating of all the working parts, the arranging of all for ready accessibility when wear or accident makes it necessary, constitute in all an intricate problem, the solution of which is well worth looking into. Nothing is final, and everything achieved is a challenge to surpass it, so that doubtless later, or even now, there may be a better machine than this we have before us, or than any we now know of, and we must note its points of excellence while we may.

There is that which is picturesque for the engineer as well as for the landscape gardener, and here, Fig. 27, is a picturesque and interesting machine, embodying a number of details of ingenious design well worth considering. It is a three-stage air-compressor designed to work to 1000 lb. gage pressure, with a free air capacity of about 50 cu. ft. per minute. It is a power-driven machine, the type of power application being according to circumstances. As here shown it has a pulley for a belt drive, but it may also be

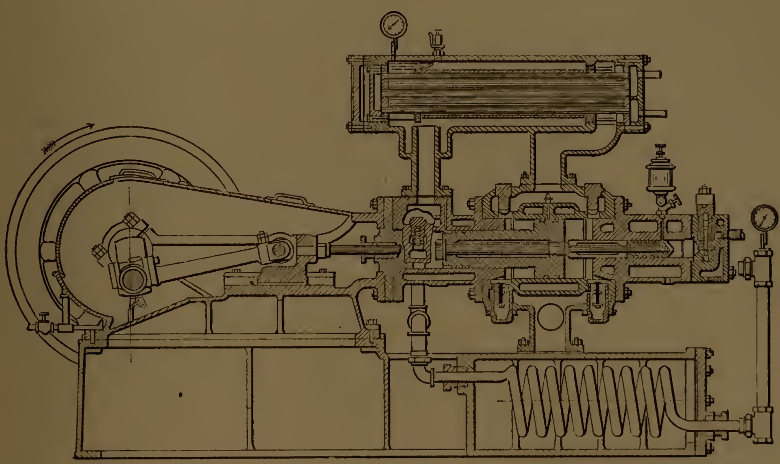

FIG. 27.—Section of Three-stage Belt Driven Compressor.

driven by a chain, by gearing, a Pelton wheel on an extension of the shaft, or by a direct-connected electric motor.

The dimensions of the cylinders are 8-, 5- and 2 3/8-in. diameter, respectively, by 8-in. stroke, and the normal speed is 150 r.p.m., giving a piston speed of 200 feet. The low pressure or intake cylinder is double acting and the other two are single acting. The three cylinders are in a straight axial line, one piston rod extending from the cross head through all the pistons. The low-pressure cylinder is between the other two, the intermediate cylinder being in front, or toward the crank, and the high-pressure cylinder at the back. The pistons of the intermediate and of the high-pressure cylinder are, in fact, plungers on each side of the low-pressure piston. The working area of the low pressure piston on each side is therefore, although we call it double acting, only that portion of the piston which surrounds the plungers, and these areas are quite different on the two sides of the piston on account of the different plunger diameters. This arrangement gets rid of all cylinder heads or partitions and stuffing-boxes between the cylinders. The only stuffing-box for the entire machine is that in the head next to the cross head and opposed to the working pressure of the intermediate cylinder. The piston rings are the only packings required besides this one stuffing-box. The packing rings for the low-pressure and the intermediate pistons are sprung into grooves turned in the solid metal. In the high-pressure piston or plunger the grooves are not turned in the solid but are formed by removable sections which fit the piston-rod and also the cylinder bore, and which are cut away at the outer corner to form the grooves for the rings. When the main portion of this piston is in place in the cylinder, a packing ring is slipped in to fit against it; one of the movable sections of the piston is then slipped in against this, then another ring, then another piston section with a washer and nut outside, which secures all. The middle or low-pressure piston has a taper fit on the rod, and is secured by a nut outside the intermediate plunger, which thus locates it precisely and holds it securely.

The actual working clearances of one of these compressors, as determined by the inspector, were: For the low-pressure cylinder, 3/32 in. and 3/32 in.; intermediate cylinder, 1/8 in.; high pressure, 1/8 in.

The distribution of the pressures and the apportionment of the work of compression throughout the cycle of operations of this compressor are such as to make the work for each stroke nearly the same. On the forward stroke, or with the pistons moving toward the crank, the low-pressure piston and the intermediate piston are both compressing to their respective pressures, and the second intermediate pressure against the high-pressure piston is assisting; that is, its thrust, at practically constant pressure, is to be deducted from the total work done by the other two pistons.

On the backward stroke the low-pressure and the high-pressure pistons are compressing, and the first intermediate pressure against the intermediate piston is assisting in the work. By calculation, based on the assumption that the compression is adiabatic, it is found that the horse-power required for the forward compression stroke of the low-pressure cylinder is 2.79, and for the same stroke of the intermediate cylinder, 8.52 h.p. The total horse-power resistance for the forward stroke is $8.52+2.79=11.31$ h.p. The intake pressure against the high-pressure plunger which is approximately constant, is sufficient to develop 3.13 h.p. which is to be deducted from the working horse-power of the other two cylinders, giving us as the net horse-power for the forward stroke $11.31-3.13=8.18$ h.p. For the back stroke of the low-pressure piston the horse-power resistance equals 4.17 h.p., which is greater than for the forward stroke, because of the increased piston area. The resistance for the working stroke of the high-pressure cylinder equals 8.93 h.p., giving a total for both the low- and high-pressure cylinders of $4.17+8.93=13.10$ h.p. From this it is necessary to deduct the power due to the intake pressure against the intermediate piston, which is found to be 3.13 h.p. The net power for the back stroke then will be $13.10-3.13=9.97$ h.p.

While the work of the two strokes is so nearly equal, that of the back stroke is the larger, which is as it should be, as this occurs on the thrust of the connecting-rod instead of on the pull. A sufficient reduction of the terminal delivery pressure would equalize the work for the alternate strokes.

As before stated, adiabatic compression is assumed in each cylinder, with efficient intercooling between the stages. The cylinders are all very completely water-jacketed, so that the temperatures of the working surfaces are kept down and satisfactory lubrication is maintained, but there is little cooling effect upon the body of air in the cylinders during the operation of compression.

The circulation of the cooling water is accomplished by a single continuous flow through both intercoolers and all the water-jackets, there being no places where the water can remain without change. The efficiency of the intercooling is sufficiently indicated by the fact that the temperature of the air leaving the second intercooler and entering the last compressing cylinder was found upon a prolonged test to be the same as that of the initial intake, or 70°, and the temperature of the air as finally delivered, there being no aftercooler, was 188°. The final temperature for perfect adiabatic, single-stage compression to 1000 lb. would be above 1250°, which would be prohibitive in practice.

In a test which was made of one of these machines, in which the number of strokes of the compressor required to fill a receiver of known capacity was ascertained, the volumetric efficiency of the compressor, or the ratio of the volume actually delivered to the total piston displacement of the low-pressure cylinder, was determined to be 0.927.

This compressor considered in detail will be found to be an extremely simple one when the complication of function is allowed for. The main frame combines in itself all the particulars upon which perfect and maintained alignment depends. The first cylinder has a true seat against the frame and the other cylinders successively locate themselves by their seating, all the joints being scraped and the faces going together without packing. Every valve is independently accessible by the removal of its cap. Taking off the high-pressure cylinder (which is no more work than the removal of a cylinder head) makes all the pistons easily removable, and gives complete access to all the cylinder interiors.

For four-stage compression the problem of distributing and equalizing the load is a simpler one than for three-stage compression, and it is not necessary to consider it here.

CHAPTER IX

AIR-COMPRESSOR REGULATING DEVICES

The most numerous class of air-compressors in the world, and in a class by themselves, are the so-called air-brake pumps on steam locomotives, these being automatically started and stopped by slight variations of the air-receiver pressure. In the same way the small electric-driven compressors employed in connection with the air brakes on elevated, subway and surface cars are automatically controlled, and, similarly to these, small isolated electric-driven compressors distributed along the line of the New York subway and elsewhere for switch and signal service, are now entirely responsive to the fluctuations of the air consumption. As these compressors, from the nature of the service are required to be in operation only a small portion of the total time, and at intervals which cannot be predetermined, it is proper that they should be thus stopped and started.

The stopping and starting of these as the pressure rises or falls comprises their entire control, and there is no variation in the rate of compression when running. Air-compressors proper, which we are chiefly to consider here, are in larger units than these, and are designed for nearly continuous service, and the operating and control of them is a different matter. The problem of air compression as a whole, as applied to what we may call regular or standard compressors, is by no means as simple as would at first appear. In most particulars it is in sharp contrast to that of the pumping of water, which may be said to offer almost ideal conditions as to power adjustment and the disposal of the output. The head against which pumping is done, as, say, by waterworks pumps, is usually constant, the resistance is uniform throughout the pumping stroke, the cylinder is always entirely full of water at both ends, so that clearance has not to be reckoned with, and the work of pumping is usually steadily continuous, there being generally ample storage reservoirs for the water, so that the pump can run right along all day, and night as well as day, always adjusted to its most economical or otherwise most satisfactory gait.

In the case of the air-compressor, its delivery pressure is, indeed, expected to be constant, provided that the air is not used at a rate exceeding the capacity of the machine, but all the other conditions are as different as could be imagined. While the terminal or delivery pressure is constant the resistance to be overcome by the piston for each stroke begins at nothing and gradually increases until full delivery pressure is reached, which then continues uniform for the last quarter or third of the stroke. There is always a certain clearance-space unswept by the piston at the end of the stroke, this diminishing to that extent the delivery volume, and this undischarged air by its re-expansion giving a preliminary shove to the piston at the beginning of its return stroke. The air is almost never used at a uniform rate, while the capacity of the compressor should be sufficient for the maximum demand, which would of course imply, if the compressor ran steadily, an excess of air at intervals, while the air storage capacity provided is always necessarily small as compared with the output of the machine, so that in practice it is quite necessary that the compression and delivery of the air shall be controllably variable within a considerable range of capacity, instead of continuing uniformly at the maximum.

Safety Valve and Throttle Control.—If such control of output is not provided, and if the compressor is run right along at full speed and capacity, the only thing to do is to trust to the safety valve on the air receiver to dispose of the surplus, a wasteful practice which no one could be expected to approve of, and which few would allow as a frequent occurrence, preferring rather the irksome task of watching the gage and manipulating the throttle, and then if the machine could not be run slowly enough of letting it stop entirely, with usually the work of barring over the center to start up again.

Automatic Speed Regulation.—On steam-driven machines all has been done that could be done to regulate the output by automatically varying the speed. This has in the nature of the case never alone been satisfactory because the range of possibilities is so limited. On straight line or single-cylinder machines the control of speed alone is especially unsatisfactory because the machines can never be run very slowly with the load on, and they stop on the centers most ignominiously. Duplex machines can of course be run much slower without stalling, but speed control is in any case coming less and less to be employed, especially because so many machines driven by electric motors or by water-power direct are necessarily run at constant speed.

Compressor builders have exercised their ingenuity and have provided devices various and innumerable for reducing the air output of the compressor without allowing it to stop, or indeed without changing the speed at all. These have been successively described and advocated in builders' catalogues, which descriptions it is not our function here to reproduce, but some of the devices may be mentioned to show the progress which has been made in this desirable feature of control.

The Run Around.—A device which has been used upon a great many single-stage steam-driven compressors provides, when a predetermined pressure has been reached, for automatically opening one or more of the discharge valves in each end of the air-cylinder at the same time, thus allowing the air at delivery pressure to play back and forth from one end of the cylinder to the other, the pressure against the opposite sides of the piston being thus for the time balanced, and the power used considerably reduced, while the delivery of air stops entirely. When the unloading occurs it is provided that at the same time the flow of steam at the throttle is choked off, so that the crank shaft will keep on turning at about normal speed ready to resume work when the reverse movement of the unloader occurs. The fact that by the action of this device the work is either all on or all off, instead of gradually increasing or reducing the load according to the demand, causes the unloading and the resumption to occur more frequently than is desirable, and the friction of the already heated air in playing back and forth through restricted passages must raise its temperature still more and cause trouble in that direction.

Choking the Intake.—In the choking intake arrangement all the intake air for both ends of the cylinder comes to it through a single pipe, and a valve is placed here for choking or throttling the intake according to the air consumption, the valve being automatically adjusted by the pressure in the receiver. Normally the intake pipe is fully open, and when the receiver pressure rises this pressure operates the valve, closing it partially or entirely as may be. The operation of the valve is gradual and there is no shock or suddenness in either the choking or the reopening of the passage. The partial vacuum created by the choking of course adds to the power required to drive the piston as compared to what it would be if running perfectly free.

The Skip Valve.—When the choking controller is applied to the intake of the first or low-pressure cylinder of a two-stage compressor and the air is wholly or partially choked off by it, the high-pressure cylinder still continuing to work at full capacity, the pressure in the intercooler is thereby abnormally reduced and the air passing through it, not having been sufficiently compressed and heated cannot receive its proper cooling effect, and also more than its share of the work of compression is thrown upon the high-pressure cylinder with an abnormally high temperature for the air finally delivered. To correct this skip valves, are provided at each end of the high-pressure cylinder to allow the air to by-pass automatically from this cylinder back to the intercooler when the pressure in the latter has been reduced to a point to which the valves are adjusted. This arrangement equalizes the work of the two cylinders and keeps the temperature of compression down.

Inlet Valve Step by Step Regulation.—On a power-driven, duplex, tandem, two-stage air-compressor; that is, a machine with both a low-pressure and a high-pressure cylinder on each side, air regulation has been accomplished by the successive automatic release or closure of the inlet valves in successive pairs, one high pressure and one low pressure at a time. As there are four such pairs of valves, the machine is thrown out of operation one-quarter of its capacity at a time, while the power required though proportionally reduced remains equitably distributed for the shaft rotation. All the inlet valves are of the Corliss type with releasing gear which does not release when the cylinder is normally compressing, the valves opening to give free admission for the entire intake stroke, and closing just before the compression stroke begins. The action of the unloader for either valve is to admit air to a trip cylinder which pushes out a plunger that in turn operates a trip cam and releases the hook. The valve then remains open and the air plays freely in and out of the cylinder, atmospheric air for the low-pressure cylinder and intercooler air for the high-pressure cylinder. There being eight Corliss inlet valves they are unloaded in the following order:

First stage: left-hand low-pressure front end and left-hand high-pressure rear end.

Second stage: right-hand low-pressure front end and right-hand high-pressure rear end.

Third stage: left-hand low-pressure rear end and left-hand high-pressure front end.

Fourth stage: right-hand low-pressure rear end and right-hand high-pressure front end.

This gives the following variations of capacity:

Full-load, 3/4 load, 1/2 load, 1/4 load, no-load. At whatever stage of release the machine is working the high-pressure and low-pressure cylinder ratio is the same, and whatever air is being delivered is compressed at full two-stage economy. The throwing off of a quarter of the load at a time does not produce any perceptible rush of current in an alternating-current motor, while to throw off suddenly the entire load of a large compressor is apt to produce surges in the line. An additional advantage of this method of unloading is that at the same time that it reduces the power consumption it also reduces the pressure on the journals; when the capacity is reduced one-half, the load on the bearings also is reduced to one-half, and when there is no-load on the cylinders there is no working load on the bearings.

Controlled Inlet Valve Closing.—On compressors driven at constant speed, and with Corliss inlet valves provided with Corliss trip valve gear, it is a natural and obvious suggestion to control this trip by the varying air-pressure, making the release and closure of the inlet valves to occur sooner or later at varying points of the stroke and upon the ends of the cylinders in pairs, in this way securing a graduated control embracing the entire range of compressor capacity.

Inlet Valve Opening.—Instead of closing the intake valves, and so reducing the volume of air taken into the cylinders, which air after admission is compressed and delivered, another way is to automatically open the inlet valves and keep them open either for the entire compression stroke or for any portion which may be necessary to sufficiently restrict the compression. As long as any inlet valve remains open the air is free to play in and out of the cylinder, and while it is thus free to come and go there is of course no compression; also only a small portion of the power required for compression will be called for, and as the air is constantly changing it does not overheat the operating parts.

Clearance Controllers.—Among the latest devices put into practical use for controlling the volume of air compressed and delivered, especially by compressors driven at constant speed, by direct-connected electric motor or otherwise, is the clearance controller. In the use of this neither the intake nor the discharge valves are interfered with in any way. There is simply an enlargement of the clearance space in each end of the cylinder, or, if two-stage, in both ends of both cylinders, so that the air which is not required for delivery into the system is merely compressed to the required pressure, and then upon the return stroke of the piston is allowed to re-expand against the retreating piston and give back the power by which it was compressed. By this arrangement the additional advantage is that the tempera-

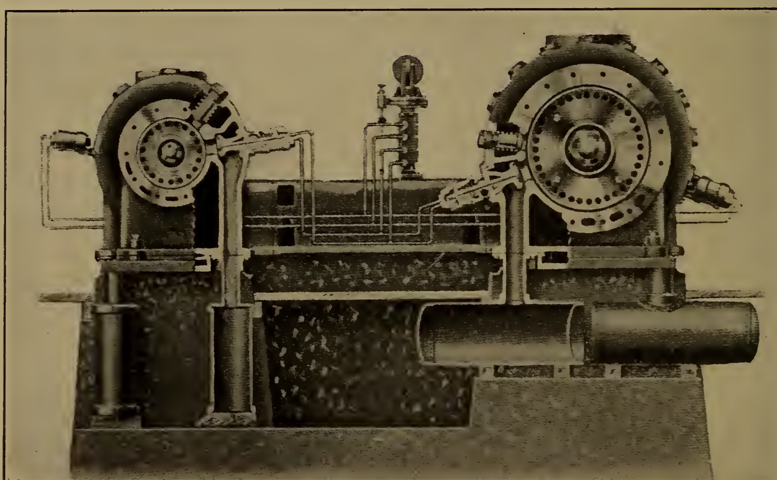

Fig. 28.—Section of Two-stage Compressor with Clearance Controller.

ture of the air is not augmented by the operation, for although the usual rise of temperature occurs on the compression stroke there is a corresponding fall of temperature upon the return stroke, and the water-jacket effect is also continuously operative.

Connected by large passages with each end of each cylinder are two clearance pockets the combined capacity of which, with the passages, is just sufficient to contain the entire cylinderful of air when compressed to the working pressure without delivering any of it, or, preferably, delivering a minute portion only. The valves which open or close the communication between these clearance pockets and their respective cylinders are controlled by slight variations of the air receiver pressure either way from that which the regulating device is adjusted for, and it is so arranged that these valves shall open or close in successive pairs instead of simultaneously, so that change of air delivery will be made fractionally instead of completely. All the valves for all these auxiliary clearance pockets are normally closed, and the compressor works and delivers the air up to its full capacity until some reduction of the delivery is required.

On either end of either cylinder, if only one of the clearance pockets is opened then only that is added to the normal clearance and one-half of the cylinderful of air will be delivered to the air receiver, while if both passages are opened then none of the air for that stroke will be discharged. The operation of this system of regulation may be assumed in each case to begin with the low-pressure or intake cylinder, but when one or both of the clearance pockets are opened for either end of that cylinder similar pockets are opened for the corresponding end of the high-pressure cylinder.

The half-tone, Fig. 28, shows an end view partially sectionalized, of a two-stage compressor equipped with this style of clearance control, and in Fig. 29 we have a series of indicator cards showing the operation of unloading, beginning with full delivery and ending with no delivery.

The cylinders in this case are respectively 27 1/4 in. and 16 1/4 in. in diameter by 24-in. stroke, driven by a direct-connected motor at a constant speed of 150 r.p.m. The cards for the low-pressure cylinder are on the left-hand side of the cut, and those of the high-pressure cylinder on the right hand, the scale of the former being 24, and of the latter 60, although all the cards have been reduced (but equally) in the reproduction. The special data for each operation accompany the corresponding cards.

At the top are typical cards familiar to all showing full delivery, with the too familiar choppy discharge lines which no one seems able to eliminate.

In the second pair of cards one end of each cylinder is still working at full capacity while the other end of each is half unloaded, or the total air delivery is reduced one-quarter.

In the next pair both ends of each cylinder are half unloaded, discharging only one-half their contents for each stroke, or reducing the total delivery one-half.

In the fourth pair one end of each cylinder is half unloaded and the other end is entirely unloaded, reducing the normal delivery three-quarters, and the machine actually delivering only one-quarter of its total capacity.

In the lower pair of cards both ends of both cylinders are entirely unloaded and no air is being delivered, or so little that it is not in evidence.

Throughout this entire series of regulating changes nothing

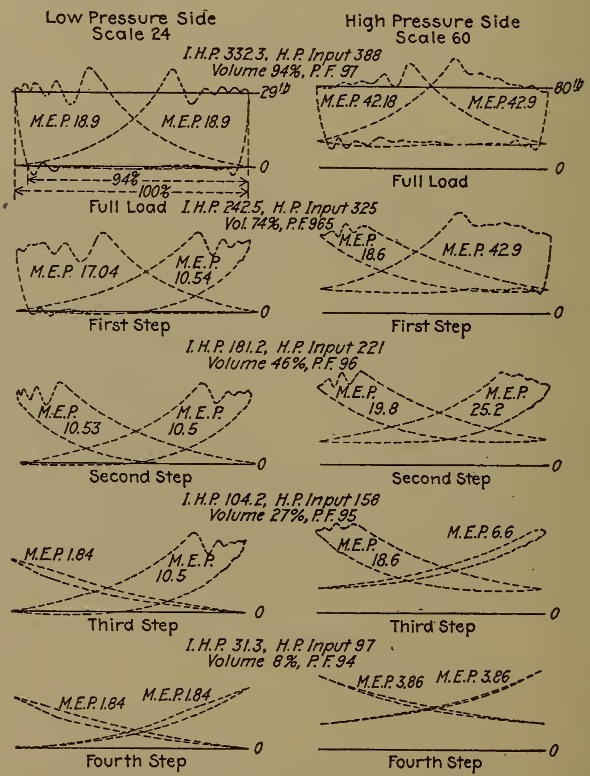

FIG. 29.—Indicator Cards Showing Action of Clearance Controller.

has been done in any way to either the inlet or the discharge valves, and they are ready instantly to resume their functions when the clearance pockets are automatically shut off, partially or entirely, which will occur when the receiver pressure is lowered below the point of adjustment. The thinness of the lower cards tells the eye at once how little is the power required when no air is being delivered.

The principle of compressor control by the use of clearance chambers, with means of opening or closing connection to the compressing cylinder, as described above, has suggested the use instead of a single clearance chamber for each end of each compressing cylinder and in full communication with it. This chamber, preferably a cylinder, would be of such capacity when empty as to contain the entire content of the compression cylinder after compression to the delivery pressure, and in that case no air would be delivered, but would be alternately compressed and re-expanded with each stroke of the compressing piston. Then if this clearance cylinder were fitted with a piston which could be automatically advanced or retracted, by the action of the fluctuating pressure in the receiver, the clearance capacity could be so varied as to change the delivery all the way from full capacity to nothing at all, the full delivery occurring when the clearance piston had been advanced to its limit and the clearance reduced to nothing.

CHAPTER X

THE DRIVE OF THE COMPRESSOR

It is not necessary here to give descriptions of the present modes of driving air-compressors or to question and discuss what drive is best. This will depend upon what means of driving are available in the given case, but in a general way it holds true that this work like any other should be done as cheaply as possible. Having made all reasonable arrangements for economically compressing the air, whatever can be saved in the cost of the drive should be sought with equal avidity and would equally result in proportionate profit.

And first as to the steam-driven machine. There was a time when two-stage compression could be strenuously advocated by a builder—which was all right—with a possible power saving of 10 per cent. or at the most 15 per cent., while at the steam end of the same machine the power cost might be from 50 per cent. to 200 per cent. more than it need be. This condition has at last been quite generally recognized, and the straight-line, steam-driven machines with plain slide valves, little or no cut-off, low steam-pressure, no condensation, etc., have given place to the Corliss (or equivalent) high-pressure compound condensing machines with all the possible incidental steam economics in addition to those at the air end.

Changes in compressed-air practice, or, more precisely, in air-compressing practice, have been succeeding each other with astonishing rapidity since the present century began, and nowhere more noticeably than in New York City and its immediate vicinity. Here compressed air has found a great opportunity, and the great contractors, seemingly begotten by the occasion, have proved themselves quick learners and then apt exemplars. The North River and the East River tunnels and the earlier subway work not only provided the ways and means for what is best in compressed-air practice, but the magnitude of the undertakings made it worth while, and even imperative, to adopt the

Fig. 30.—High Falls Compressor Plant Rondout Siphon Contract.

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most reliable and economical apparatus procurable, for the compressing regardless of the cost of installation.

Accordingly the several air-compressing plants employed by the great contracting companies for the work here spoken of were regarded as models of up-to-dateness in every detail which could promise economy and precision of working, and both the contractors who owned and operated them and the designers and builders who were their sponsors, all were proud of them, and the technical press wrote appreciative descriptions of them as a labor of love.

These were steam-driven plants, with all the accepted steam and power economizing devices, which need not here be enumerated in detail, and the air ends also provided for two-stage compression with efficient intercooling and automatic regulation, which allowed no power to run to waste on account of the variations of the load.

Although the work for which these compressors were employed was necessarily of a temporary character there was no temporizing or makeshift about them, and day by day, comparing their performance with that of "simple" machines which might have done the work, they piled up the savings and paid for themselves over and over.

They made such an excellent showing that, humanly speaking, we might have said that these machines, when their first job was done, would be sure of employment again as soon as there was more work in their line to be done. They were still practically as good as new, so there could be no plea of old age suggested against them.

The first batch of New York river tunnels, the first subways, the Pennsylvania Railroad terminal were all completed by their aid. Then came the great water tunnel to be driven the entire length of Manhattan and across Brooklyn to Staten Island, and new subways without end to be built, but for all this later and closely following work not one of these compressors of such excellent record came into play.

Electricity Jumps in.—While these fine compressors were at work and making such an admirable record other things also were working. Especially was electricity having a phenomenal business development. Splendidly equipped light and power plants of almost unlimited capacity were being installed in New York, as well as in every large city, and these, taking advantage of every possible economy in steam generating and in power development, were enabled to offer current for driving air compressors and for other power purposes at rates which practically compelled acceptance.

The electric drive under such circumstances is an attractive proposition. The first cost of the electric-driven compressor, including the motor, is much less than that of the steam-driven plant, with boiler, condenser, piping and other appurtenances, and fuel handling and storage arrangements. The operating force required is reduced to one-fourth or less and the ground occupied is also minimized, with no exacting requirements as to location, so that generally the compressor may be placed much nearer the work, with a great saving in the pipe lines. The power required is always ready without preliminary notice and the power cost stops entirely when the compressor stops. The advantage of the electric drive thus offered has been so evident from a strictly business and money making viewpoint that it could not be ignored, and now electric-driven plants have been installed for the extensive new lines of work, because it was simply the cheapest thing to do.

What may perhaps be considered the culmination of the development of the contractors' steam-driven air-compressing plant was that at High Falls, New York, for the Rondout siphon and adjacent work of the Catskill aqueduct system, Fig. 30. This was said to be the largest installation of high-pressure steam-driven compressors in the world. It especially emphasized the fact that when it comes to economical working the improvements and the savings have been much greater in the steam-drive than in the compressing apparatus. However apparently temporary may be the character of the work, it is found to pay to install all the devices and arrangements of economizing function which would belong to the most permanent lines of employment, and accordingly here all the familiar steam economizing devices were employed, such as high-pressure steam—150 lb.—compound steam cylinders with condensers, feed water heaters, economizers, etc.

It is customary to think only of the saving of fuel in such a case as this, but the labor saving also cuts a figure which quite compels notice. Here were ten large duplex Corliss, compound steam, two-stage air machines running day and night up to full capacity, developing over 4000 h.p. and delivering 24,000 cu. ft. of free air per minute compressed to 110 lb., using

100 to 110 tons of "birdseye" coal per day, with hand firing; and yet the entire operating force, including engineer-in-charge, was only eight men for each 8-hr. shift.

Nevertheless, and notwithstanding the success of this plant, if the selecting and installing of it had come only five years later we may believe that it would have been a very different plant, because by that time the electric companies would have been submitting proposals for supplying current.

In Fig. 31 we get a glimpse of the compressing plant (installed 1912) of a large contractor for the building of the Lexington Avenue subway, New York. It is located at 96th Street and First Avenue, close to the East River, and it may be regarded as typical of the plants employed upon this line of work at this time, there being at the time of this writing a score of compressors of this type in operation in Greater New York.

This plant comprises five electrically driven cross-compound compressors with cylinders 25 1/4 and 15 1/4 in. in diameter and 21 in. stroke, delivering air at 100 lb. with an individual free-air capacity at 187 r.p.m. of 2110 cu. ft. per minute or an aggregate of 10,550 cu. ft.

These machines have direct connected self-starting synchronous motors with belted exciters, 365 b.h.p., 6600 volts, 3 phase, 25 cycle, the rotor mounted on the crank shaft with a bearing in each frame, this arrangement giving a greater efficiency than that of a high-speed belted motor. The rotor is heavy enough to give a sufficient flywheel effect, in this assisted also by the pulley which drives the exciter. A speed of 187 r.p.m.—654-ft. piston speed—is high for a compressor of this size, bu it proved practicable and safe with this type of machine.

A special conduit running lengthwise of the building is provided for the free air supply, the piston inlet providing a ready means of connecting. A large intercooler is set transversely above the cylinders of each machine, forming the air connection between them. In this case special cooling tubes of "admiralty bronze" are provided, so that the more or less saline water of the East River may be used for the circulation. The air pressure at the intercooler is 27 lb. After leaving the intercooler and before entering the high-pressure cylinder the air passes through a separator which takes care of all the water liberated in the air by the intercooling.

The compressor runs at constant speed, the air output being regulated by an automatic clearance controller spoken of more in detail in the preceding chapter. Looking over one of these machines in detail it will easily appear that a modern air compressor of the highest type is by no means so simple a machine as some might easily imagine, but all the refinements have full business warrant for their being.

From the compressor house the air is carried by a 10-in. pipe westward to Lexington Avenue, about half a mile, and then it is led away both north and south long and increasing distances through 8-in. pipes and smaller. The 10-in. pipe from the compressor house is laid in the gutter along 96th Street close to the curb and carried overhead wherc it crosses three avenues. In the middle of each block a sliding expansion joint is placed, and the long vertical pipes and curves at the crossings also yield more or less. As the air starts on its journey its temperature is not less than 200°, but before Lexington Avenue is reached not the slightest trace of heat remains, which shows how quickly the air cools off, and the futility of ever attempting air reheating except close to where the air is to do its work.

From the nature of the work in this case it is impossible to make any intelligible comparison between the power used at the compressors and the power actually realized where the work is going on, as that is so widely distributed, so intermittent and of such varied character. The only thing until the work is completed is to have enough air at effective pressure always ready.

New York Water Tunnel.—The driving of the great water tunnel, in solid rock all the way, from Hill View Reservoir outside the northern boundary of Greater New York, across the Borough of the Bronx, under the Harlem River, then the entire length of Manhattan, under the East River, across the Borough of Brooklyn, under the Narrows and then to Silver Lake in Staten Island, the tunnel varying in diameter from 15 ft. to 11 ft., is being done by sinking shafts, say half a mile apart, to depths ranging from 200 ft. to 750 ft. below the surface and then driving the tunnel in each direction until a continuous tube results. The drive for this work is one or two isolated electric-driven compressors at each shaft, these calling for no special comment.

The electric drive for the air compressor is now common enough in every part of the world where the transmission of power from distant waterfalls can be made profitable. Such

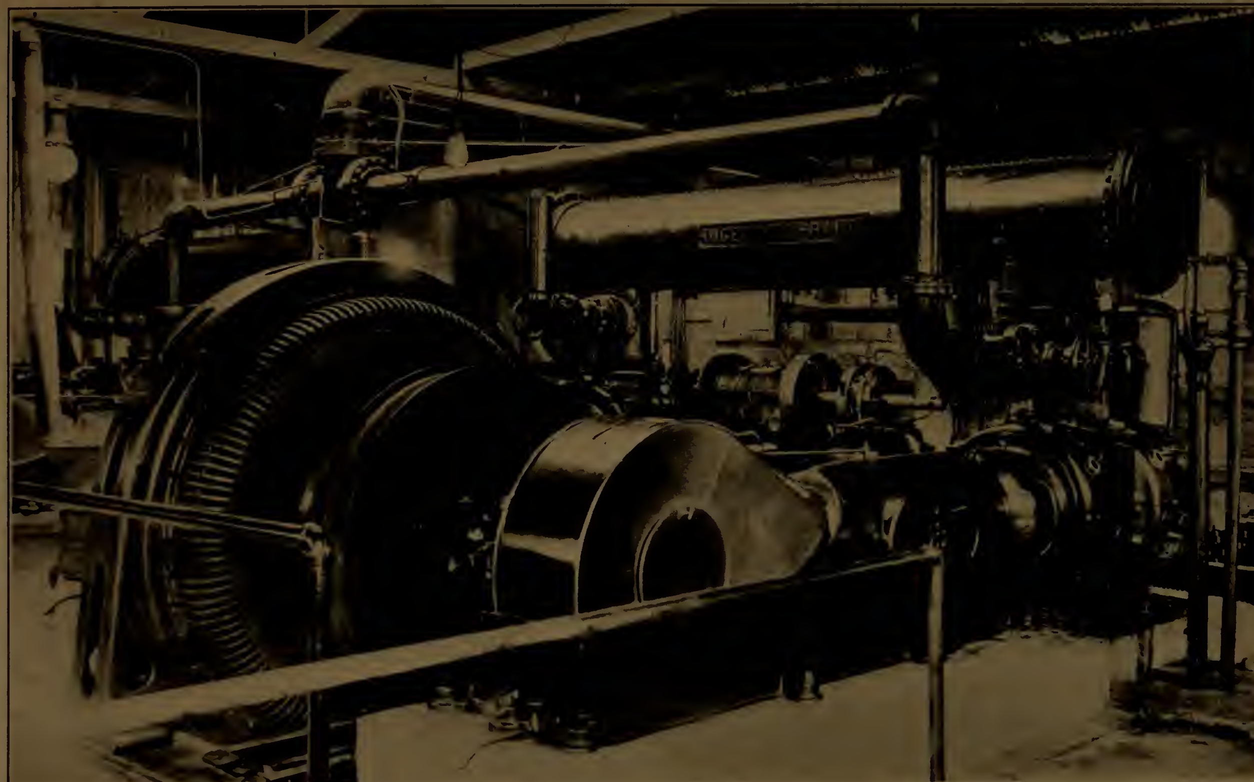

FIG. 31.—Electric Driven Compressors—Lexington Avenue Subway.

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drives are now generally preferred and take precedence of the direct water-driven compressor, especially on account of the saving in the length of pipe lines usually effected. There have been in the not distant past a number of interesting installations in which a Pilton water wheel has been mounted directly upon the compressor crank shaft, and these have usually given good satisfaction, but that was before the electric drive had attained its present development, and to-day the Pelton wheel which would have been driving a compressor direct is much more likely to be driving a generator instead for the doing of the same work after the transmission.

CHAPTER XI

THE TURBO-COMPRESSOR

The fan blower held its own for so many years, and with so little change in design, so little advance in scope and efficiency, that it almost seemed a finality in its special field. Those who are in the habit of accepting things as they are might for many years have considered the fan blower as the one perfected or finished mechanical device, not that its performance was so satisfactory, but that there seemed so little promise of improving it.

Its work for so many years was the blowing of blacksmith fires and foundry cupolas, with also a lot of ventilation service and some conveying of shavings and other light materials, but it generally was only worked up to, say, 1 lb. pressure. If higher pressures were required the so-called "pressure blowers," such as the Root or the Connersville, came into play, but even they did not like to undertake pressures above 4 or 5 lb.

It is singular, however, that the pressure possibilities of centrifugal blowers were in sight, at least of the theorists, long ago. It was only a question of speeds, the pressure increasing as the square of the velocity, and no one had practically approached the limit. Table XII (here somewhat changed in form) appeared in Appleton's *Applied Mechanics* in 1878. The velocities here given, it is to be noticed, are air velocities and not fan velocities, but there are builders of fans to-day who claim and who graphically demonstrate that the air velocities are greater than the peripheral speed of their fan blades. The table in the latter portion of it could be only theoretical as it seems to go beyond the permissible speed limit. This limit is supposed to be nearly reached in some steam turbines which have a peripheral velocity as high as 1300 ft. per second.

The highest air velocity in the table, by the way, is 1016 miles per hour. In the United States meteorological records a wind velocity of 100 miles an hour is recorded as having been reached just once. The pressures being as the squares of the velocities,

TABLE XII.—AIR PRESSURES AND VELOCITIES FROM CENTRIFUGAL ACTION

Pressure, lb. per sq. in.	Air velocities, ft. per sec.	Pressure, lb. per sq. in.	Air velocities, ft. per sec.
0.625	83	2	471
0.125	116	2.5	527
0.25	166	3	577
0.375	204	4	666
0.5	236	6	816
0.625	263	8	943
0.75	288	10	1,053
0.875	312	12	1,154
1.0	333	15	1,300
1.5	408	20	1,490

only our imagination can suggest what would be the force one hundred times as great as that of a hundred-mile wind.

In all these centrifugal machines—fans, blowers or compressors—the responsible element, that which fixes the speed limits and the pressure possibilities of the individual units, is the rotor. Looking at the rotors of some of the standard types of fans or blowers, Fig. 32, they seem to be well designed to fly to pieces if run fast enough, and there is a sharp contrast between them and

Fig. 32.—Typical Rotors of Fan Blowers.

the rotors of turbo-compressors, as Fig. 33, but these also have their speed limits.

The rotor or impeller in the turbo-compressor as thus far developed is generally not left to assume the entire responsibility for the compression. It contains vanes which give the air the rotation velocity of the impeller before leaving it, but stationary vanes are also usually provided for deflecting the air and converting its velocity into pressure. In Fig. 34, *A*, *A*, *A* are the rotating impeller vanes and *B*, *B*, *B* are the fixed deflecting vanes.

Figs. 35 and 36 will suffice to indicate the essential features of the turbo-compressor, Fig. 35 being sections of the impeller and its vanes and Fig. 36 a vertical longitudinal section of a turbo-compressor unit with five impellers A, A, A. The air enters the first impeller as indicated by the arrow, is thrown out into

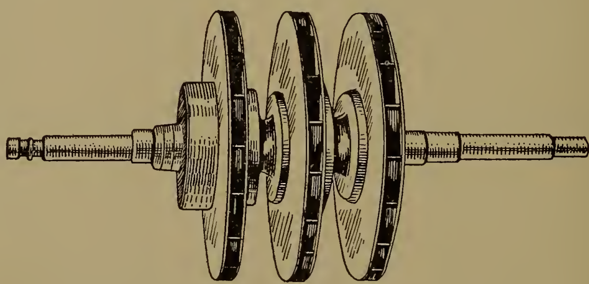

FIG. 33.—Rotor of Turbo-compressor.

the thin, flat surrounding passage B and then curves back into passage C by which it is led to enter the second impeller to repeat the cycle as before, and so on. As the air in passage B reaches the circumference of the shell, still with a high but somewhat diminished rotative velocity, its centrifugal force would oppose

FIG. 34.—Impelling and Deflecting Vanes.

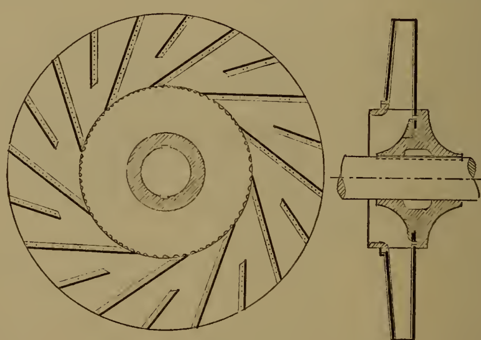

FIG. 35.—Sections of Impeller.

its return toward the center, and accordingly at the turn and in passage C there are fixed deflecting blades which intercept the air, change its direction and convert its centrifugal force into a considerably increased pressure with which it enters the second impeller.

The spaces *D, D, D* and *E, E, E* are filled with cold water in constant circulation, and the air passing in comparatively thin sheets between these cool surfaces has its temperature much reduced. Indeed, except when entering and passing through the impeller the air is constantly in contact with these water-cooled surfaces and the compression in each successive impeller being not sufficient in that alone to raise the temperature very high the air emerges from the series at comparatively low temperature. When the compression is to high pressures, two or three such series as here shown are employed, and the air in passing from

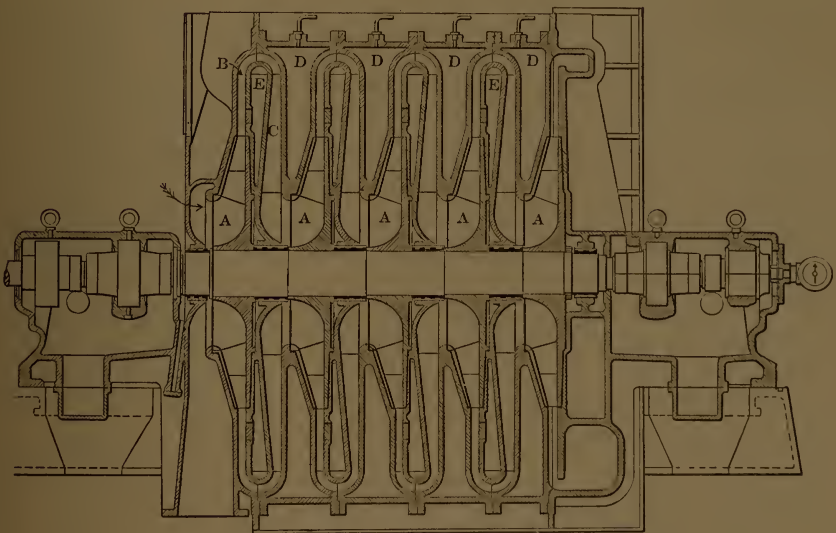

Fig. 36.—Vertical Section of Turbo-compressor.

one to the next is carried through an efficient intercooler, and throughout the entire compression the temperatures are kept so low that the compression is nearer isothermal than with any reciprocating machine.

As early as 1906 there was published a report of the performance of a Rateau turbo-compressor designed to compress 1716 cu. ft. of free air per minute to 7.2 atmospheres. At a speed of 4500 turns per minute, which gave the required pressure, the output was 25 per cent. greater in volume than demanded, and on increasing the speed the pressure was raised another atmosphere. This compressor was in four distinct parts, each consisting of a group of

32 impellers in series upon one shaft, with intercoolers between the groups. In the first group the pressure was raised from 1 to 1.7 atmosphere; in the second to 2.9 atmospheres; third, to 4.9 atmospheres and in the fourth to 7.2 atmospheres. The ratios of absolute pressures before and after each compression were thus successively 0.5882, 0.5862, 0.5918 and 0.6805, which showed a remarkably equitable distribution of load for a type of apparatus still novel.

When we look at Fig. 36 a question naturally arises: The five successive impellers as here shown are of the same size, and, being upon the same shaft, must have the same volumetric capacity, while the actual volume of air entering each successive impeller must necessarily be smaller, on account of the immediately preceding compression. This evidently is not as it should be, and it is not easy to see how the later impellers of the series find their proper share of employment or are able to accomplish what they should in the way of additional compression. The case is apparently not very different from what it would be in a two-stage or a three-stage reciprocating machine if all the compressing cylinders were of the same capacity.

The simplest way suggested, and in fact carried into successful practice, for correcting this is to make each successive impeller thinner than the preceding, or of less volumetric capacity in proportion to the reduced volume of air which it is to receive from its predecessor. Only the impeller capacities require successive reduction, that of the connecting passages having no bearing upon the results after the machine is in full operation.

It does not appear that changing the diameters of the successive impellers, which has been done, could be a satisfactory solution. A reduction of peripheral speed would be a reduction of compressing force, and, in correct proportion to the volume to be compressed, this is as much required for the last impeller of the series as for the first if all are to be equally efficient.

It is not within the scope of the present publication to go into details either of design or of construction. Although the essential principle of the turbo-compressor is extremely simple there are many essential conditions to be complied with. The shaft and entire rotor system must be perfectly balanced or the high rotative speeds are impossible. Although except the journals there are no surfaces to be accurately fitted to work metal to metal they must still be adjusted to run very close to each other, and this proximity may lead to actual running contact and disastrous consequences if the expansions and contractions due to temperature changes are not provided for. The rush of the air and the working pressures developed in the running cause thrusts in the direction of the axis, which if they cannot be balanced one against the other must be otherwise taken care of.

Although the practical value and the high efficiency of the turbo-compressor have been demonstrated beyond all question, and although many machines are in successful and established operation the device as a whole cannot be regarded as beyond the experimental stage, and the detailed descriptions of perfected machines, as perfection goes in such lines, still belong to the future.

Air-compressor history has been made with great rapidity in the last twenty years or so, and even now it may be that another chapter as important as any is being added. It is less than a quarter of a century since the present writer first came into personal contact with a practical, commercial air-compressor of the period. It was for such air-pressures as were then demanded for the running of rock drills employed in subterranean work, and there was then little demand for compressed air for any other purpose. The crude and uneconomical compressors first accepted were generally superseded after too many years by steam-driven machines which no one had any reason to be ashamed of, and now these have been driven out of employment by the electric drive, as spoken of in the preceding chapter.

Now what if, when the reciprocating compressor, whatever may be the drive of it, has been brought to its highest state of efficiency and reliability, it is to be quite generally superseded? This seems to be the almost universal law of inventional progress. We get all that we can out of a given device, explore and develop all its possibilities, begin to think it nearly perfect in its line, and then find it crowded out of service when at its best by some other thing which is just beginning its course triumphs, it also to be knocked out later by some as yet unknown successor.

There can be little room for doubt as to the great future awaiting the turbo-compressor, especially for the larger units. There are actually being built at this writing single turbo-compressors with capacities of 50,000 to 70,000 cu. ft. of free air per minute delivered at a pressure of 3 atmospheres, and in smaller units, but still exceeding the capacities of the largest reciprocating machines, that are capable of delivering air at pressures above 10 atmospheres

The two great power companies on the Rand, South Africa, have at present in operation six steam-driven turbo-compressors of 4650 h.p. each and six electric driven of 4300 h.p. There are also building for these companies three steam-driven turbo-compressors of 9500 h.p. each, the aggregate of these being 82,200 h.p. and the air is delivered at pressures suitable for driving rock drills, hoists, pumps and general mining service.

The turbo-compressor seems to offer the ideal conditions for direct driving by either steam turbine, electric motor or Pelton wheel, although the latter has as yet made little record. The ultimate efficiency compares well with that of the best reciprocating machines. Satisfactory means of control is provided either for constant volume with varying pressure or for constant pressure and a fluctuating rate of consumption. The special adaptation of the turbo-compressor seems to be in the furnishing of a constant flow of air for blast-furnaces.

A Mechanical Wonder.—One of these machines which has taken the place and does the work of a large Corliss, cross-compound, condensing, two-stage air machine, maintaining a constant supply of air at 90 lb. for the various uses of a large manufacturing establishment, I am inclined to regard as in one particular the greatest mechanical wonder of the century; and that is in its absolute noiselessness. While it is working along steadily, at full speed and pressure, any one standing near it fails to discover by the ordinary evidence of the senses, by eye or ear or by the touch of the hand, that there is any life in it, and only by going around the end of the machine and watching the running of a worm wheel which has to do with the governing apparatus can any assurance of motion and work be obtained.

An important particular to be noted in connection with the employment of the turbo-compressor for general compressed-air service is the condition of the air delivered. It at no time gets very hot and is finally discharged at temperatures as low as that of the output of any other type of compressor. The air as delivered has its usual accompaniment of moisture, which may be got rid of by the usual separating and draining devices, so that the air when used is as dry as when compressed by any other means. The air is delivered clean and without any trace of oil to be deposited in receiver and piping, so that with the turbo-compressor the too familiar ignitions and explosions will be impossible.

It seems to be the general impression that the turbo-compressor is to find employment only or chiefly in exceptionally large units, that in capacity it is to be and to remain the Jumbo among compressors. The present writer is under the impression that this may not be so. What if at the present time the builders of centrifugal air-compressing apparatus may be all on a still hunt for the smaller or general compressor trade, and all on the quiet expecting to knock big holes in the reciprocating compressor business? Quite recently a personal letter was addressed to three firms understood to be most prominent builders of apparatus in these lines requesting them to kindly send what printed matter they were known to be giving out relating to centrifugal blowers or compressors for pressures above or considerably above those of the ancient fan blower which are measured by inches of water. The letters were courteously answered and in each case there was sent precisely what was not asked for, which carried not a particle of information of the character desired, and what was wanted was extracted from these parties by getting business friends to pose as possible customers to whom the printed matter wanted was forwarded by return mail.

Those who are afraid that competitors will find out something should read Kipling:

> "They copied all I could follow,
> But they couldn't copy my mind;
> And I left 'em swearing and stealing
> A year and a half behind."

There may be a very different explanation of the situation. The apparent reticence of builders as to turbo-compressor developments at the present time may be because none of them have achieved such success that they are able to brag about it.

CHAPTER XII

THE TAYLOR COMPRESSOR—THE HUMPHREY PUMP

While we are considering the various means and methods of compressing air for industrial purposes we cannot omit some notice of the Taylor compressor. This is a perfectly practical device which has had but a few actual installations, but always with complete success.

The essential principle of the Taylor compressor was known to the ancients. The general proposition as embodied in the Taylor development would seem to be somewhat paradoxical. Given a decided fall of water in a rapidly flowing stream, no matter what may be the actual height of the fall, and it may be made to compress air to any pressure desired.

And yet there is not the slightest suggestion of "perpetual motion" or of "something for nothing" about it. The work done, represented by the quantity of air compressed to the given pressure, will always be less by a certain percentage of inefficiency than the potential energy of the water which has flowed to do the work.

It is not easy to determine the precise mechanical efficiency of this device although it is claimed to have been done, and figured statements of efficiencies are matters of record; but it happens that in this case the matter is not important, for it seldom can happen that the flow of the stream where it may be installed will be constant and that the "compressor" will use exactly all of it.

The Taylor air compressor is a device quite similar to what we might assume the air lift to be with its operation reversed, both being dependent for their action upon the diffusion of air through a moving column of water, in the latter case the expansion of the compressed air serving to raise the water while in the former device the fall of the water compresses the air.

The most notable Taylor compressor installation is that on the Montreal River at Ragged Chutes, near Cobalt, Ontario, Canada. There was here a drop in the river of 54 ft. within a quarter of a mile, the stream being capable of furnishing 5500 h.p. and the "compressor" was designed with a computed capacity of 40,000 cu. ft. of free air per minute compressed to a gage pressure of 120 lb., thus making the compressor at the moment easily the largest single compressing unit in the world. The location was within reachable distance of a mining location capable of using all the air that could be furnished. It is transmitted through 9 miles of 20-in. pipe, from the end of

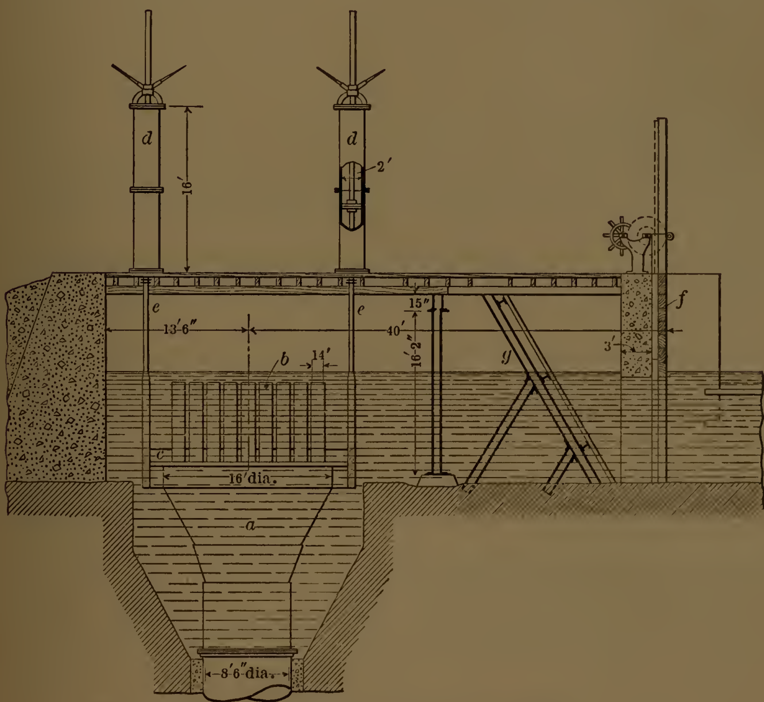

FIG. 37.—Head of Taylor Compressor.

which there are two 12-in. branch pipe lines. About 7 miles from the compressor there is another 12-in. branch, making the total length of main piping about 21 miles.

The compressing operation will appear in the progress of the description of this interesting plant condensed from the authoritative account of Mr. C. H. Taylor written for *Mines and Minerals*. The water is controlled at the summit level by suitable gates. After passing the gates the water flows through two 16-ft. diameter intake heads, one of which is shown in Fig. 37 at *a*. In each of these heads there are 66 pipes, *b*, 14 in., in diameter set in a steel disk *c*. Below these open pipes the heads gradually diminish in diameter until they become 8 ft. 4 3/4 in., and from this point they are 15 ft. long. These pipes telescope into the intake shafts which are 8 ft. 6 in. in diameter and 345 ft. deep, the orifice of the head being at the surface of the water. This arrangement permits the heads to be raised or lowered, to conform to the level of the water in the forebay, or the heads may be

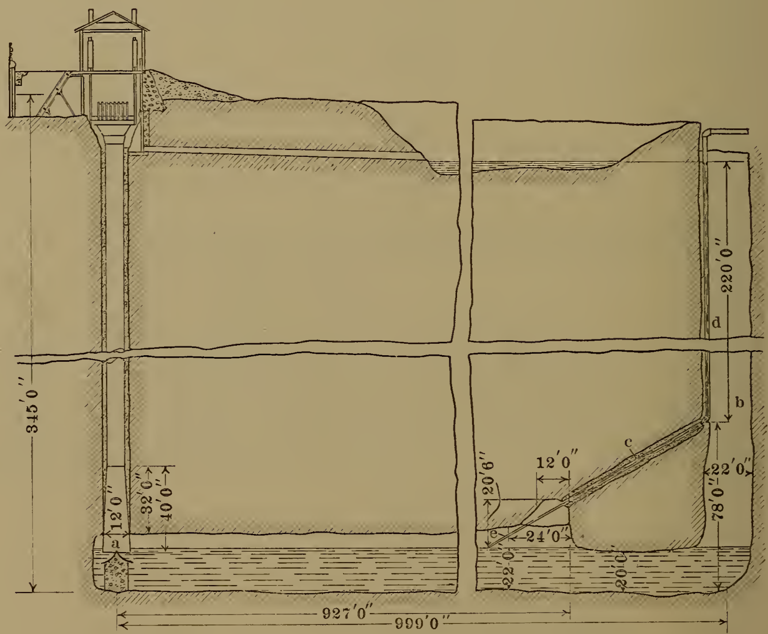

Fig. 38.—Complete Section of Taylor Compressor.

raised above the level of the water by air hoists, *d*, thus cutting off the supply entirely. The two air-hoist cylinders *d* act as governors, automatically raising and lowering the heads which are suspended from them by the hangers, *e*, thereby regulating the flow of water into the intake pipes *b* according to the demand.

The water, with the entrained air which has been drawn in through the pipes *b*, flows through the heads with a descending velocity of from 15 to 19 ft. per second, this velocity gradually decreasing, owing to the compression of the globules of air, and finally there is a further reduction in velocity owing to the enlargement of the last 40 ft. of the shaft, as seen in Fig. 38. By the time the water reaches and strikes the steel-capped concrete diverting cones *a*, its velocity is so diminished by the baffle from the compressed air that there is little shock. The increase of velocity which gravity would cause in falling through the shaft does not occur because the force of gravity is overcome by the pressure of the air in the chamber or tunnel below.

The cones *a* are for the purpose of spreading the flow of air and water, thereby bringing the air nearer the flow through the tunnel. The air being lighter than the water, it rises to the surface of the water, the pressure here being 120 lb. per square inch. This tunnel is 20 ft. wide, 26 ft. high and 1000 ft. long in the down-stream direction, this unusual length being for the purpose of utilizing the total head of the stream, the length not being necessary so far as the separating of the air from the water was concerned, before the latter started up the outlet shaft *b*. As the velocity of the water is only about 3 ft. per second, practically all of the air leaves the water in the first 300 ft.

The pressure given to the air is due to the height of the body of solid water in the outlet shaft, which in this case is 298 ft. deep and 22 ft. in diameter, and the air-pressure is therefore constant. If the level of the tunnel was higher, making both the inlet and the outlet shafts shorter the air-pressure would be proportionately lighter, and *vice versa*. The water flows along the tunnel and up the outlet to the river, the difference in elevation between the mouths of the intake and the discharge tunnels being 47 ft., this difference in height causing the flow of the water, notwithstanding that the air-loaded water in the down shaft is specifically lighter than that in the up shaft.

Near the outlet end of the tunnel its height is increased to 42 ft., and at this place two pipes are carried through the 30 degree riser *c* to the uptake shaft. One pipe *d*, 24 in. in diameter, carries the compressed air to the surface, where it is connected with the 20-in. main air-pipe line. The other pipe *e* is 12 in. in diameter and has its lower end submerged at a safe distance above the roof of the outlet portion of the tunnel, to act as a blow-off in case the air in the tunnel should acquire such pressure and volume as to force the water below the level of the tunnel outlet. If the air were allowed to escape up the outlet, instead of being carried up by the pipe, it would lighten the column of water in that shaft and the air-pressure would not be constant.

The blow-off pipe ends at the upper level of the water in the outlet shaft, its end remaining open to the atmosphere. When the volume of the air is greater than the demand the air accumulates in the upper part of the tunnel, forcing the water down and exposing the lower end of the blow-off pipe *e* to the compressed air, thus allowing a portion of the water in this pipe to drop back, thereby reducing the weight of the remaining water in the pipe to less than the pressure of the air. The equilibrium is now overcome and the water in the pipes is driven upward to the surface, where a most spectacular sight is witnessed, as the body of water is shot out by the air sometimes to a height of 500 ft. The blow-off continues until the pressure of the air in the tunnel is sufficiently reduced to allow the water to again submerge the end of the pipe. Water now rises until an equilibrium is established between the air- and the water-pressure in the tunnel. The air-pipe and the blow-off pipe are packed in concrete the entire length of the 30-degree riser in order to seal them in and prevent any escape of air up the outlet shaft.

It will be seen that the initial cost of the plant at Cobalt, or of any similar plant, is not a small one. A writer, apparently well informed but perhaps not any too friendly, when the installation was approaching completion wrote as follows:

"The air is conducted to Cobalt by a 20-in. pipe. The pipe was made in 40-ft. lengths with welded flanges, and sliding expansion joints set in concrete pits every half mile. Aside from the transmission pipe lines I would estimate the cost of the plant at the Chutes as not far from $1,000,000. This makes the cost per horse-power (5000) about $200, which does not compare favorably with the cost of an ordinary air plant. I know of two small plants that were installed for less than $90 per horse-power including flume and pipe line, as well as wheel and compressor."

The entire compression of the air by this means is almost absolutely isothermal, as the air in the down shaft must at all times be at the same temperature as the water with which it is intermingled. The air delivered, notwithstanding the water contact, must be "dry" air, or air in which there will not be sufficient moisture to manifest itself in the subsequent use of the air if while it is at its highest pressure and normal temperature there is proper drainage of the main pipe line by which the air is transmitted. The air also is free from oil, so that there will be no possibility of ignitions or explosions in receivers or elsewhere, and the air would be almost the ideal, pure, fresh air for miner's use except for one unanticipated condition which has developed. It is found that the air delivered has less than its normal proportion of oxygen.

It was at once found that it was practically impossible to burn candles in the mines when supplied with air from the Taylor compressor. A sufficient explanation was furnished at once when analysis determined that the air contained only 17.7 per cent. of oxygen instead of the normal 21 per cent. The lack of oxygen does not apparently trouble the miners, but besides the difficulty experienced in keeping lights, the effect of the gases from exploded dynamite is much quicker and more serious than was found to be the case with air compressed by the usual machinery.

The oxygen is abstracted from the air by the water with which it is in immediate contact during compression, and it appears that this result is precisely what should have been expected. This matter is quite convincingly explained by Prof. Olin H. Landreth, Dean of Engineering, Union College.

The loss of oxygen, he writes, in hydraulic air compression is due to the well-known fact that water will absorb different amounts of different gases, just as it dissolves different weights of different soluble materials.

TABLE XIII.—PERCENTAGE OF GASES ABSORBED TO VOLUME OF ABSORBING WATER-PRESSURE, ONE ATMOSPHERE

Temperature deg. C.	Oxygen, per cent.[1]	Nitrogen, per cent.
0	1.027	1.856
5	0.891	1.630
10	0.787	1.450
15	0.704	1.307
20	0.635	1.191
25	0.575	1.096

Absorption of Oxygen and Nitrogen in Water.—For oxygen and nitrogen, Table XIII gives the approximate percentage by volume absorbed to the absorbing water. This is stated for normal pressure, or one atmosphere; for other pressures the weights of oxygen absorbed increase in approximately direct proportion to the pressure, but as the density also increases as

[1] These are true percentages; ratios are 1/100 of the above percentages.

the pressure, the volumes (as measured under the varying pressure) are about constant.

The figures given in the absorption table are for atmospheric oxygen and nitrogen, not single, separate gases. They therefore represent conditions just as one would find them in hydraulic compression. The table shows that water absorbs from 80 to 90 per cent. more of nitrogen from the atmosphere than of oxygen.

OXYGEN ABSORBED MORE READILY THAN NITROGEN

If, however, these tabular percentages be divided by 0.21 for the oxygen and by 0.79 for the nitrogen, being the proportions of each gas in 1 cu. ft. of air, it is seen that for the same volume of gas offered for absorption, oxygen is absorbed more than twice as fast as nitrogen, and the composition of the air taken up by the water is richer in oxygen than the original air. As the absorbed air is mostly carried away by the water flowing from the uptake shaft, the air which remains and which is used for industrial purposes is poorer in oxygen than the original air. The absorbed air is largely given up from the water, but only after the water has risen in the uptake till the pressure is reduced, and even then is largely held in minute globules which give the water a milky appearance.

The Humphrey Pump and Compressor.—The changes which are occurring in the means and methods of power development are having their effect upon the work of air compression and wider departures from established practice are likely to follow. While at the beginning the larger air-compressors were quite generally steam-driven, many now are operated by gas- or oil-engines, saying nothing here of direct and indirect drives by water-power and by electricity however generated. The oil-engine working mechanically, with reciprocating pistons and rotating shafts, is finding a rival for pumping service in the Humphrey pump which dispenses with the mechanical devices and yet shows the highest efficiencies, using completely all the expansive force of the successive charges of explosive mixture. Then, as an afterthought or a following up of the invention, the column of water set in motion by the explosion, instead of having the further end of it driven into a water delivery pipe, is made to expend its force in compressing in a suitable chamber, and then expelling into the compressed-air system, a volume of air commensurate with the disposable force.

It is not easy to tell the story of the Humphrey pump in few words. The following is mostly abstracted from a paper by its inventor, Mr. H. A. Humphrey, before the Manchester Association of Engineers:

The simplest form of Humphrey pump is shown in Fig. 39. Imagine a charge of gas and air to be compressed in the top of chamber *C* and fired by a sparking plug projecting through the top casting. All valves are closed when the explosion occurs and the increase in pressure drives the water downward in *C*, setting the whole column of water in the discharge pipe *D* in motion. The column attains kinetic energy while work is being done on it by the expanding gases, and may move with considerable velocity when these have reached atmospheric pressure. The motion of the water column cannot be suddenly arrested, hence

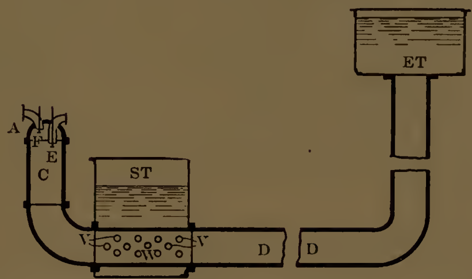

Fig. 39.—Simplest Type of Humphrey Pump.

the pressure in the combustion chamber *C* tends to fall below that of the atmosphere, the exhaust valve *E* opens, and also the water valves *VV* in the supply tank *ST*. Water rushes in through *VV* mostly to follow the moving column in pipe *D*, but partly to rise in *C* in an effort to reach the same level inside the chamber as exists in *ST*.

When the kinetic energy of the moving column has expended itself by forcing water into the high-level tank *ET* it comes to rest, and there being nothing to prevent a return flow, the column starts to move back toward the pump, and gains velocity in the return direction until the water reaches the level of the exhaust valve *E*, which it shuts by impact. A certain quantity of burnt products is now imprisoned in the cushion space *F*, and the energy of the moving column is expended in compressing this gas cushion to a greater pressure than that due to the static head of the water in tank *ET*. Hence a second outward movement of the column results, and when the water reaches the level of valve E the pressure of the space F is again atmospheric, and further movement of the water opens valve A against a light spring, and draws in a fresh charge of gas and air. If there were no friction, the water would fall to the same level as that from which the last upward motion started, but the amount of combustible charge drawn in is slightly less than this movement would represent. Once more the column of water returns under the elevated tank pressure, and compresses the charge of gas and air, which is then ignited to start a fresh cycle of operations.

The action of the pump is not altered if, instead of delivering into an elevated tank, it discharges into an air vessel, or into an open-top standpipe or tower, and both these arrangements are useful if a continuous flow from the outlet is desired.

In the simple form of pump the degree of compression of the combustible charge prior to ignition depends on the height to which the water is raised, and exceeds the static equivalent of the head. This is obvious if one remembers that the kinetic energy acquired by the liquid column on its return flow is utilized in compressing the combustible gas, while the compression brings the column to rest.

The same considerations enter into the question of the cushion pressure attained, but here we are dealing with the compression of a volume of gaseous fluid which occupies the clearance-space only, and the stroke of the water column is greater in proportion, because for the first part of the stroke exhaust products are being expelled, and no compression occurs. The cushion pressure rises rapidly as the height to which the water is lifted increases, and at the maximum lift of about 40 ft. to which the simplest type of pump is limited, it may exceed the explosion pressure when using producer-gas, or may approximately equal the explosion pressure when working with city gas or gasoline.

A Humphrey pump was exhibited in operation at the Brussels International Exhibition, and obtained two "highest possible" awards—namely, a Grand Prix in the class for gas-engines, and a Grand Prix in the class for pumps. Its construction is similar to that in Fig. 39, and explanation is not needed till we come to the valve gear shown in Figs. 40 and 41. It will be observed that a bolt B sliding horizontally must lock either the admission-valve A or the exhaust-valve E by engaging under collars a or e, which are fixed on the stems of their respective valves. Now the bolt is urged right or left, according to whether spring s_1 or s_2 is pulling the hardest, and this again

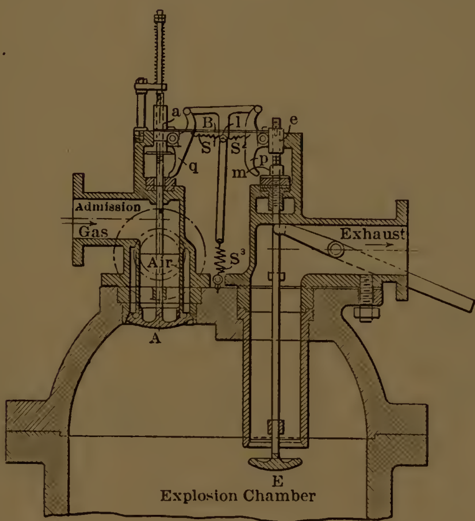

FIG. 40.

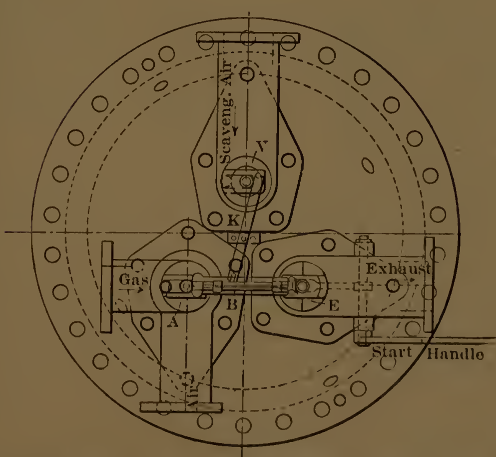

FIG. 41.—Vertical Section and Plan Showing Valve Gear of Humphrey Pump.

depends on whether the link l, to which the springs are attached, has been shifted to the right or left. Suppose the exhaust-valve opened last, then its washer m, engaging against cam arm *p*, moves the system *p*, *l*, *q*, so that it leans to the right, in which position it is retained by the tension of spring s_3. This puts tension on spring s_1 and loosens spring s_2; bolt *B*, therefore, tries to move to the right, but until the exhaust valve shuts it can only press upon collar *e*. However, when valve *E* comes on its seat the bolt instantly locks under *e*, and the same motion which holds valve *E* shut has released valve *A*, so that next time a suction occurs in the combustion-chamber, *A* only can open. Precisely the same kind of action occurs when *A* shuts and is locked and *E* is released again. Thus valves *A* and *E* are automatically allowed to act alternately, the difference between

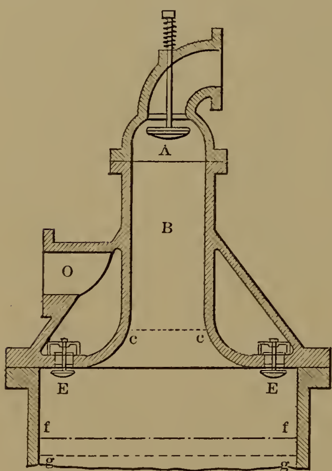

Fig. 42.

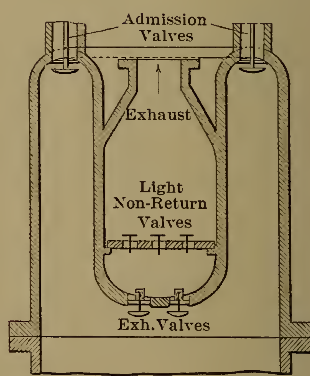

Fig. 43.

Figs. 42–43.—Alternative Arrangements of Two-cycle Pump Without Valve Gear.

them being that while *E* remains open till shut by the rising water, *A* shuts under the action of its supporting spring, so soon as the suction in the chamber permits the spring to lift the valve to its seat.

The scavenging-valve *V* is shown in the plan of the combustion-head, Fig. 41, and as it operates at the end of each expansion stroke, its locking and release periods correspond with those of the exhaust-valve, and are made simultaneous by a lever pivoted at *K*, and operated by a pin on the bolt *B*. If the water could rush in fast enough when the pressure falls to atmosphere, there would be no scavenging action; but the incoming water has to be accelerated, and that just gives rise to a sufficient suction to effect the desired scavenging. In the exhaust outlet there is a light non-return valve, to prevent burnt products being drawn back into the chamber.

Figs. 42 and 43 show alternative arrangements of the top of the combustion chamber for a two-cycle pump requiring no valve gear. The combustion chamber has to be specially shaped, so that the incoming charge, which may be preceded by pure air, displaces the burnt products and mixes as little as possible with them. Thus, in Fig. 42, *A* is the admission valve at the top of the tall, narrow part of the chamber *B*, in which the full charge volume extends down to the level *cc*. *A* number of exhaust valves *E* lead to a common exhaust outlet *O*, which may be fitted with a non-return valve, or each exhaust valve may carry a light non-return valve on its spindle, as shown. The level at which expansion reaches atmospheric pressures is, say, *ff*, but this level having been reached by the water, its further movement draws in fresh combustible mixture till it occupies the space down to *cc*, and the liquid level has fallen to *gg*. The column of liquid now returns and drives the exhaust products through the valves *E*—which had opened by their own weight—until these valves are shut by the water. The kinetic energy acquired by the column is now spent in compressing the fresh charge, which is ignited to start a new cycle. Thus, each outstroke is a working stroke, and no locking gear is required on the valves.

The same cycle applies to Fig. 43, but in this case there is a series of admission valves placed in a ring so as to allow the mixture to enter with a low velocity in order to prevent eddies and mixing with the exhaust products. A higher compression pressure is obtained with this pump than with the simple pump, and consequently higher efficiencies with the same lift.

Fig. 44 shows the arrangement of a double-barrel pump, which has two combustion chambers, *A* and *B*, in which explosion occurs alternately. Any Humphrey pump, whether single or double barrel, may be converted into a high-lift pump by means of an air-vessel fitted with valves, and called an "intensifier." The idea is to first allow the water-column to gain velocity, and then to utilize its kinetic energy to (a) compress an elastic fluid, and (b) deliver water under the pressure to which the elastic fluid has been compressed.

In Fig. 44, *A* and *B* are the barrels of a two-barrel pump, and at the end of the play-pipe *D* there are two air-vessels *E* and *F*, the latter being large enough to give a continuous flow at outlet *O*, and to maintain a practically uniform pressure. The smaller air-vessel *E* is fitted with a downwardly projecting pipe *K*, open to the atmosphere at the top, and carrying a valve *L* at its lower extremity, arranged to close under the action of the rising water. The cycle starts with explosion, all valves except *L* being shut, and the water level as shown. While the water level in *E* is rising to *L*, air is merely being discharged into the atmosphere, and as no work is being done by the column of water, it gains speed until valve *L* is shut by impact. Imprisoned in *E* there is now a definite quantity of air, which suffers compression until its pressure reaches that at which the high-pressure water-valves *W* can open, and allow the remaining kinetic energy of the column to force water into *F*. Valves *W*

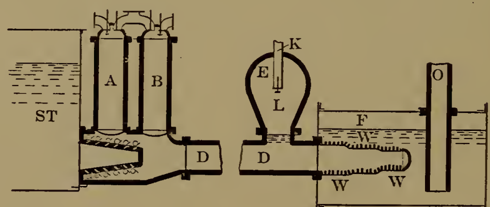

Fig. 44.—Double-barrel Pump with Two Combustion Chambers.

close when the column comes to rest, but there remains enough energy in the compressed air in *E* to give, by expansion, the return flow, which causes exhaust in *A* and compression of the fresh charge in *B* to start a fresh cycle. When the water level falls below valve *L* this valve opens, and air is admitted into *E* for the rest of the return stroke.

Now it is easy to see that if the pipe *K* is made vertically adjustable with regard to *E*, the point of the cycle at which *L* shuts can be varied, and more or less air entrapped in *E* at will. But the amount of energy stored in this air will also vary with its quantity, for we assume that the degree of compression remains constant, and is indeed fixed by the pressure maintained in *F*. Consequently the ratio of the total energy of the working stroke to the energy stored in the compressed air in *E* can be made anything desired, or, in other words, we can obtain any compression pressure of the new charge in B which we like, and this independent of the water lift. The advantage is obvious, for compression pressures equal to those in modern gas-engines can be employed with a corresponding increase in thermal efficiency. Further, by manipulating the position of pipe K a given pump can be made to meet any conditions as to height of lift, for if the lift increases K can be raised, so that the energy stored in the air in E remains the same, there being now less air, but at a higher pressure.

An important development of the arrangement is shown in Fig. 41, and notice is directed to the fact that at each cycle air is drawn into, and rejected from the vessel E. Let us suppose K to be connected to a supply of combustible mixture instead of opening into the atmosphere, we shall then have an automatic pump for taking in mixture and discharging it under pressure. If the discharge is into a reservoir from which combustion chambers A and B can be supplied, we have at once a means of quickening the cycles and greatly increasing the output of a given size apparatus. It is convenient to replace vessel E by two vessels, one for air and one for gas, so as to maintain the combustible constituents separate until they enter the combustion-chambers. If the first portion of the out-stroke of the water column is allowed to reject the surplus air and gas back to the sources of supply, then the action throughout the cycle is precisely that described when using the single vessel E, except that a larger proportion of the total energy is absorbed in the compression of air and gas, but the excess is given out again during the expansion of the pre-compressed charge in either A or B. The chief advantage arises from the more rapid working, as there is no longer any need to wait for the water-level in A or B to fall under the action of gravity when the charge is being taken in. In fact, the apparatus becomes practically independent of the water-level on the supply side.

The Humphrey Pump as an Air-compressor.—If the column of water oscillating in the play-pipe of a Humphrey pump is used as a water piston, and caused to rise and fall in an air vessel fitted with suitable valves for the inlet and outlet of air, the combination constitutes an air-compressor of a very efficient type and promising many advantages. Take the case of a single-barrel pump and a single air vessel, shown in Fig. 45. The cycle, which may take two seconds to accomplish, is as follows:

	Pump chamber *A*	Air-compressor chamber *C*
1st out-stroke........	Expansion to atmosphere. Intake of scavenging air.	Expulsion of air till water shuts valve *s*. Compression and discharge of compressed air till water shuts valve g. Cushion till water comes to rest.
1st in-stroke..........	Exhaust till water shuts valve *e*. Cushion till water comes to rest.	Expansion of cushion to atmosphere. Intake of fresh air.
2d out-stroke.........	Expansion of cushion to atmosphere. Intake of combustible charge in excess.	Compression of air, but not sufficient for further delivery.
2d in-stroke..........	Rejection of surplus charge till water shuts valve *r*. Compression of charge till water comes to rest.	Expansion of compressed air.

The flexibility of the air-compressor can now be studied. To begin with the pump side, the level of the inlet valve *e*, and the rejected charge *r*, are assumed to be variable, although Fig. 45 being merely a diagram, does not show how the pipes carrying these valves are moved vertically. As the level of

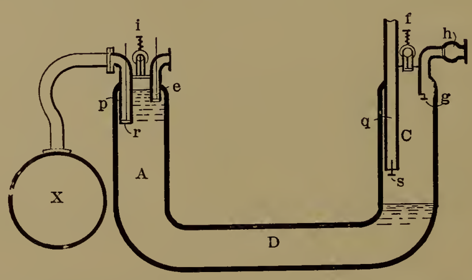

Fig. 45.—Humphry Pump an an Air Compressor.

these two valves controls the amount of charge ignited at each cycle, and the amount of the cushion space, their regulation is all that is required to increase or diminish the energy developed per working stroke. On the compressor side the position of the valves *g* and *s* controls the cycle of operations on this side of the apparatus, and renders it possible to compress a large volume of air to a low pressure, or a smaller volume of air to a high pressure, or to make any intermediate changes which may be desired.

Thus all the conditions of output, up to the full limit of the compressor, may be governed at will, and for all ranges the compression pressure of the new charge may be kept up to the required degree, so that the apparatus works at its maximum efficiency throughout the whole range. The amount of water which oscillates between the chambers should theoretically be altered along with the total capacity per working cycle, but the reason for this is merely to prevent the last portion of each down stroke from being wasted by taking in surplus combustible mixture in one chamber, or surplus air in the other chamber, to an undue extent. If the surplus of the combustible mixture is unnecessarily large, the extra amount rejected will increase the pressure in the reservoir X, and this increase of pressure may be made to automatically govern a water supply, and so bring up the total volume of reciprocating water, or to allow part of the water already in the apparatus to escape, so as just to keep a small amount of excess charge for each cycle, no matter what may be the output of the pump.

The Humphrey pump is far beyond the experimental stage. It has been built in several units up to 1000 h.p. and its economy and efficiency may be said to be fully established. Four Humphrey pumps, with a capacity of 33,000 gallons per minute each, against a head of 25 feet, have been installed and are in successful operation at Chingford, England, pumping water from the river Lee as a contribution to the water supply of the city of London.

Not much can as yet be said as to any extensive employment of the Humphrey principle to practical air compression, but it seemed at least proper to mention it here as among the practicable possibilites.

CHAPTER XIII

POWER COST OF COMPRESSED AIR

What is the power cost of a cubic foot of compressed air at any given pressure? We will not now look into all the possible economies of the case, but will try to get at the actual cost according to the simplest methods of compression.

Say, then, that we have a steam actuated air compressor, with steam-cylinder and air-cylinder both 20 in. in diameter by 24-in. stroke, running at 75 revolutions or double strokes per minute, using steam at 80 lb. and compressing air to 80 lb. The case will then be like this:

Power required by air-cylinder:

$20^2 + 0.7854 \times 36.6$ (mean effective pressure) $\times 300$ (feet per minute piston speed) $\div 33{,}000 = 104.53$ h.p.

Power required in steam-cylinder:

$$104.53 + 10 \text{ per cent.} = 114.98 \text{ h.p.}$$

Volume of free air compressed per minute:

$$20^2 \times 0.7854 \times 300 \div 144 = 654.5$$
$$654.5 - 10 \text{ per cent.} = 589 \text{ cu. ft. free air}$$

Volume of air when compressed to 80 lb.:

$$589 \times 0.1552 = 91.4 \text{ cu. ft.}$$

Power of steam-cylinder (steam 80 lb., cut-off 0.25, mean effective pressure or resistance 40.29 lb.):

$$20^2 \times 0.7854 \times 40.29 \times 300 \div 33{,}000 = 115.06 \text{ h.p.}$$

Volume of steam used:

$$20^2 + 0.7854 \times 75 \div 144 = 163.62$$
$$163.62 + 10 \text{ per cent} = 180 \text{ cu. ft.}$$

Here 180 cu. ft. of steam at 80 lb. produce 91.4 cu. ft. of air at 80 lb., or 1 cu. ft. of air at this pressure costs nearly 2 cu. ft. of steam at the same pressure.

It should be remembered that the same ratio will not necessarily hold good for other pressures. For lower air-pressures the steam will have a little more advantage, and for higher pressures it will have a little less. The mean effective resistance assumed for the air-cylinder is the theoretical resistance with no cooling of the air. In practice the actual resistance is somewhat less than this, but the difference of mean effective between the air card and the steam card, or the friction loss of the machine, is also usually more than 10 per cent., so that few of the common compressors in use will at their best give any better results than the above.

Table XIV gives the horse-power required to compress 1 cu. ft. of free air per minute to a given pressure, also the horse-power required to furnish a cubic foot of air at the given pressure; or, in other words, the power cost of the operation of air compression is exhibited both from the beginning and from the ending of it. From either standpoint the power required is given both for isothermal and for adiabatic compression, in the one case assuming that the air remains at its initial temperature during the compression, and in the other case that the air as heated by the compression is not cooled during the operation. The power required as given in the table is the theoretical power, and no allowance is made for the inevitable losses of power that occur in its actual application, and, of course, it makes no difference what may be the source of the power, or the economy with which it may be developed or applied. The power employed may be steam, with or without cut-off or condensation, water power, electricity, animal or manual power, or anything else.

When the volume of free air required to be compressed per minute is known, or the volume of air at the given pressure required to be furnished, the theoretical power required may be found by multiplying the total number of cubic feet required by the power required for 1 cu. ft., as here given. In the last column of the table, although the compression is assumed to be adiabatic, the air after delivery is supposed to have cooled to normal temperature, and to have assumed its practically available volume, so that the 1 cu. ft. of compressed air represented in column 5 is precisely the same as the 1 cu. ft. of column 4.

Compression Losses in Detail.—In the use of this table the second column, showing the power cost of isothermally compressing 1 cu. ft. of free air to the given pressure, represents the ideal

TABLE XIV.—HORSE-POWER REQUIRED TO COMPRESS AND DELIVER I CU. FT. OF FREE AIR PER MINUTE TO VARIOUS GAGE PRESSURES, ALSO THE POWER REQUIRED TO COMPRESS AND DELIVER I CU. FT. OF AIR AT THE GIVEN PRESSURE

1 Gage pressure	Compressing 1 cu. ft. of free air per minute to given pressure		Delivering 1 cu. ft. per minute of air compressed to the pressure given	
	2 Compression at constant temperature	3 Compression without cooling	4 Compression at constant temperature	5 Compression without cooling
5	0.01876	0.01963	0.02514	0.0263
10	0.03325	0.03609	0.05586	0.06399
15	0.04507	0.05022	0.09105	0.10145
20	0.05506	0.06283	0.12994	0.14829
25	0.06366	0.07422	0.17191	0.20043
30	0.0713	0.08464	0.21678	0.25734
35	0.0782	0.09425	0.26445	0.31872
40	0.084305	0.10324	0.31375	0.38422
45	0.08954	0.11166	0.36368	0.45353
50	0.09508	0.11952	0.41848	0.52605
55	0.09936	0.12702	0.47112	0.60227
60	0.10402	0.13418	0.52855	0.68181
65	0.10808	0.14028	0.58612	0.76079
70	0.11245	0.14718	0.64812	0.8483
75	0.11629	0.15373	0.70952	0.93795
80	0.11926	0.15971	0.76843	1.02906
85	0.1224	0.16555	0.83039	1.1231
90	0.12558	0.17096	0.89444	1.2176
95	0.12886	0.17629	0.96164	1.3148
100	0.13121	0.18153	1.0243	1.4171

and the unattainable, but still the only rational and natural standard of efficiency in air compression. Whatever the actual power employed may exceed the values in this column is the irrecoverable cost of compression.

In comparing the performance of a steam actuated air-compressor with this standard we shall find at least four different sources of loss in the operation of compression, all enforcing some deduction from the ideal efficiency. Few persons in dealing with compressed air recognize and make the necessary allowances and deductions for all of these sources of loss, and in consequence the efficiencies of the air compressors of the day are still generally represented to be higher than they actually are. In deploring the low ultimate efficiencies in compressed-air systems the inefficiencies are to be found lurking at the compressor end as much as at the air-motor end of the string.

The first deduction to be made is for the friction of the machine, and when indicator-cards are taken this is accurately represented by the difference in the mean effective pressures in the air- and in the steam-cylinders, assuming the areas and strokes of the cylinders to be the same. This difference is often lower than might be expected. In some large Corliss compressors, where the air-cylinders are tandem to the steam-cylinders, the piston-rod of the steam-cylinder being continued into the air-cylinder and carrying its piston, the total loss of power in the friction of the engine often ranges as low as 5 per cent., where the total friction of the same steam-engine if transmitting all of its power through its crank shaft would be as much as 10 per cent. In the familiar straight line, direct-acting air-compressors the friction may generally be assumed as taking 10 per cent. of the initial power, and it is seldom lower than that.

The second source of loss to be reckoned with is in the increase of temperature and the reduction of the weight or actual quantity of air admitted to the cylinder for compression. This loss is seldom recognized, and still more rarely made the subject of actual computation. It is difficult to determine it accurately, because it is the one detail in the cycle of operations in the compressing of air about which the indicator diagram has nothing to say. It is evident, however, that there must be some loss from this source in any case. As the air is always heated when being compressed, and at best only slightly cooled during the operation, whatever heat is given up by the air is transmitted to the cylinder surfaces, so that in continuous compression they become quite hot. Water-jacketing only partially cools these interior cylinder surfaces, and some parts of the cylinder and of the heads, with usually all of the piston, are not cooled at all by the water. The air, which when heated from any source we find to give up its heat so quickly in transmission, is also heated with equal celerity when the conditions are reversed, and it cannot pass through heated passages into a heated chamber, which the cylinder is, without being heated and increased in volume, so that a less weight or actual quantity of air is sufficient to fill the cylinder. The loss from this source may in many cases be light, but there can be little doubt that sometimes it is entitled to a deduction equal to that allowed for the friction of the machine. If air whose normal temperature is 60° is actually at 120° at the moment when compression begins, then the weight of air present is less than 90 per cent. of the same volume at its original temperature.

The third loss of power, or of efficiency, in air compression is due to the heating of the air during the compression, and to the greater force required for the compression on account of this heating. This is the one source of loss that is most generally recognized, and too often treated of as the only one. The loss in this case is represented by the percentage of excess of mean effective pressure above that required for isothermal compression. In compressing to 70 lb. the M.E.P. for isothermal compression is 26, and for adiabatic compression it is 33.73, and the mean of the two is 29.87. The excess of the adiabatic above the isothermal is 29.7 per cent., and the excess of the mean above the isothermal is still 14.85, or say 15 per cent. No compressor within my knowledge does its compression to 70 lb. with less than this 15 per cent. of loss except by devices that increase the friction of the machine, or add to the power required or to the cost of operation in some way.

The fourth source of power loss in air compression lies in the fact that while the indicator-cards show, as they do, that the M.E.P. for the compression-stroke is above the mean of the isothermal and the adiabatic pressures, or when compressing to 70 lb. more than 15 per cent. above isothermal compression, the volume of free air compressed is never a cylinderful. The figures in the formulas and in the tables are based upon the assumption that a certain volume of air is compressed, and when applied to the cylinder of a compressor, the actual capacity of the cylinder, or the net area multiplied by the stroke, is the volume represented. It is of course the fact that the volume actually compressed is always somewhat less than this. There is a loss at each end of the stroke. Compression of the air at full atmospheric pressure does not begin precisely at the beginning of the stroke, and all of the air is not expelled by the piston at the end of the stroke. It is customary with compressor-people to say that clearance in the air-cylinder at the end of the stroke does not mean loss of power, but only loss of capacity, because the power which has been expended in the compression of the air filling the clearance-space is returned to the piston by the re-expansion of the air when the piston makes its return stroke. The clearance does, however, in effect in our computations, represent an actual loss of power, or an expenditure of power without any result, because the allowance which the clearance demands is so generally ignored, and every stroke of the piston is assumed to compress and deliver free air to the full capacity of the cylinder, which it certainly never does.

In practice these four items of loss of power in compression occur in different combinations, such as 10, 10, 17, 10 = 60.5 per cent. net efficiency or 7, 2, 15, 5 = 73.6 per cent. net efficiency. It is safe to say that the ultimate efficiency never goes as high as 80 per cent., while it often goes below 60 per cent.

And after all this depreciative discourse not a word has been said about leakage, which of course would be another detail of inefficiency to be added to those enumerated. In air receivers and connections, and in piping for transmission, it is not proper to believe that leakage is unavoidable, or that it is more than infinitesimally permissible, but in the responsible working parts of air-compressors the case is different. There leakage may occur, and it would probably be quite an exceptional compressor in which there were not some leakage in discharge valves, inlet valves or piston, either of which would help to reduce the ultimate volume of air compressed and delivered. Indicator-cards are so far from intelligibly reporting these leakages as something to be corrected that such leakages may actually have the effect of producing a better card. Piston leakage would make the compression line lower and nearer to the isothermal, while discharge valve leakage has been actually claimed in specific instances to give the cylinderful of free air a higher pressure, or to make the cylinder seem to have taken in a greater mass of air ready for the beginning of the compression-stroke. We volunteer here no suggestion as to what percentage of deduction should be made for valve and piston leakage in a working compressor in good condition, but there are undoubtedly instances of compressors which have not been properly looked after where 5 to 10 per cent. would be warranted.

Air Consumption of Rock Drills and Pneumatic Hammers.— There is a curious thing to be noted in this connection. It is generally desirable to know about the rate of air consumption of rock drills, pneumatic hammers, etc., especially so that when a number of such tools are to be put into use it will be possible to estimate the compressor capacity required. The air consumption of such tools cannot be arrived at in advance by measurement and computation, and such data as we have come from records of actual running. In nearly all such records the air-compressor has been accepted as the air-meter. The customary way is to assume the free air compressed to be equal to the entire space swept by the piston, in many instances not even allowing for the space occupied by the piston rod, thus making the reading of the compressor-meter a considerable percentage too "fast," and the actual consumption of the air-driven tools greater than it actually is. The compressed-air output of the compressors being thus made to appear too high, the air consumption of the drills, etc., has also been rated too high, and one balances the other.

Throughout this chapter we have had in mind the single-stage air-cylinder with direct steam drive. Everything would of course equally apply whatever the drive might be, whether by belt or gear or by direct electric drive, the latter now becoming very frequent as a means of utilizing the force of distant waterfalls and also in the case of extensive engineering work located near the large cities where current is furnished at low cost by the big lighting companies.

Steam Consumption of Air-compressors.—Table XV here given was prepared by Mr. O. S. Shantz, M. E., then of Detroit, who died in Buffalo in 1910. The diagram, Fig. 46, plotted by the present writer, embodies everything contained in the table, and may be found more easily usable.

The table shows the weight of steam required to compress 100 cu. ft. of free air to the various gage pressures listed, either in single-stage or in two-stage compression. They are based on the various steam consumptions per indicated horse-power per hour shown at the head of each column. Adiabatic compression is assumed all through with a power difference of 10 per cent. between the steam-cylinder and the air-cylinder or cylinders.

In using the tables in a given case a steam consumption per horsepower-hour is assumed based upon an understanding of the type of steam end of the compressor, the steam pressure, cut-off, vacuum if condensing, etc. Under this assumed figure on the line opposite the required air pressure will be found the pounds of steam consumed per 100 cu. ft. of free air compressed.

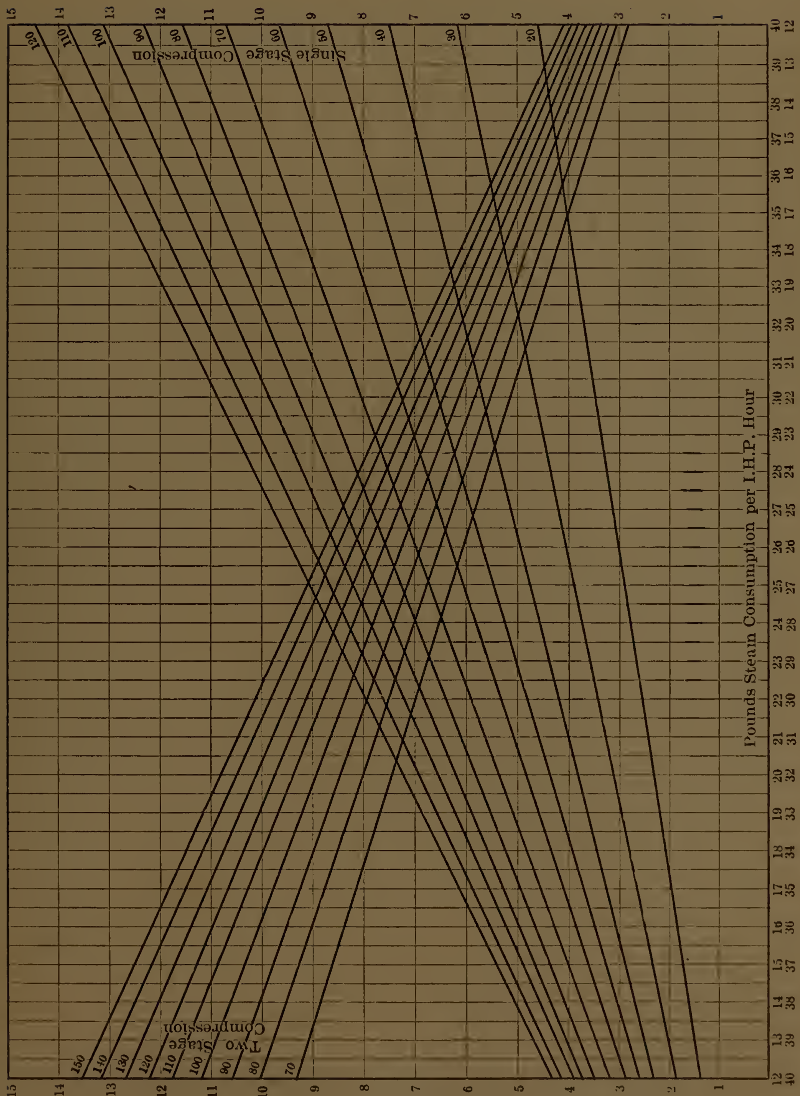

FIG. 46.—Steam Consumption of Air Compressors.

The corresponding figures at the opposite ends of the diagram indicate pounds of steam consumed in compressing 100 cu. ft. of free air to the given pressure.

TABLE XV.—STEAM CONSUMPTION OF AIR-COMPRESSORS

Single-stage Compression

Pounds steam consumption per indicated horse-power per hour	12	12½	13	13½	14	14½	15	15½	16	16½	17	17½	18	18½	19	19½	20	20½	21	21½
Air-pressure, gage 20	1.36	1.42	1.47	1.53	1.58	1.65	1.70	1.75	1.82	1.87	1.92	1.98	2.04	2.09	2.15	2.21	2.26	2.32	2.38	2.44
Air-pressure, gage 30	1.84	1.92	1.99	2.07	2.14	2.23	2.30	2.37	2.45	2.53	2.60	2.68	2.76	2.83	2.92	2.99	3.06	3.14	3.22	3.30
Air-pressure, gage 40	2.26	2.36	2.45	2.54	2.64	2.73	2.83	2.92	3.02	3.11	3.20	3.30	3.39	3.48	3.58	3.68	3.77	3.87	3.96	4.06
Air-pressure, gage 50	2.62	2.74	2.84	2.95	3.06	3.17	3.28	3.38	3.50	3.61	3.71	3.82	3.93	4.04	4.16	4.26	4.36	4.49	4.59	4.71
Air-pressure, gage 60	2.92	3.05	3.17	3.29	3.41	3.54	3.66	3.77	3.90	4.02	4.14	4.26	4.38	4.50	4.64	4.75	4.80	5.00	5.11	5.25
Air-pressure, gage 70	3.22	3.36	3.50	3.63	3.76	3.90	4.02	4.16	4.30	4.43	4.56	4.70	4.83	4.96	5.11	5.24	5.36	5.50	5.64	5.78
Air-pressure, gage 80	3.50	3.65	3.80	3.94	4.08	4.24	4.37	4.52	4.67	4.81	4.96	5.11	5.25	5.40	5.55	5.69	5.84	5.99	6.12	6.29
Air-pressure, gage 90	3.72	3.88	4.06	4.19	4.34	4.50	4.65	4.80	4.96	5.11	5.26	5.43	5.58	5.73	5.90	6.05	6.20	6.36	6.50	6.68
Air-pressure, gage 100	3.96	4.13	4.30	4.46	4.61	4.79	4.95	5.11	5.29	5.45	5.61	5.78	5.95	6.10	6.28	6.45	6.60	6.77	6.93	7.10
Air-pressure, gage 110	4.18	4.36	4.54	4.71	4.87	5.06	5.22	5.40	5.58	5.75	5.92	6.10	6.26	6.45	6.63	6.80	6.96	7.15	7.31	7.50
Air-pressure, gage 120	4.38	4.57	4.75	4.94	5.11	5.30	5.47	5.65	5.85	6.03	6.20	6.40	6.57	6.75	6.95	7.12	7.30	7.50	7.67	7.86

Single-stage Compression—(*Continued*)

Pounds steam consumption per indicated horse-power per hour	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
Air-pressure, gage 20	2.49	2.60	2.72	2.83	2.94	3.06	3.17	3.28	3.40	3.50	3.61	3.74	3.85	3.97	4.08	4.18	4.30	4.42	4.54
Air-pressure, gage 30	3.37	3.52	3.68	3.82	3.98	4.14	4.29	4.44	4.60	4.75	4.90	5.06	5.20	5.36	5.51	5.66	5.81	5.97	6.12
Air-pressure, gage 40	4.15	4.34	4.52	4.70	4.90	5.08	5.26	5.46	5.65	5.85	6.03	6.22	6.40	6.60	6.78	6.95	7.14	7.35	7.50
Air-pressure, gage 50	4.80	5.02	5.25	5.45	5.68	5.90	6.10	6.33	6.55	6.76	7.00	7.20	7.42	7.65	8.86	8.05	8.27	8.51	8.71
Air-pressure, gage 60	5.36	5.60	5.85	6.08	6.32	6.56	6.80	7.05	7.30	7.55	7.80	8.03	8.27	8.52	8.76	8.99	9.22	9.50	9.71
Air-pressure, gage 70	5.90	6.16	6.45	6.70	6.97	7.25	7.50	7.77	8.05	8.32	8.60	8.85	9.12	9.40	9.66	9.91	10.16	10.45	10.70
Air-pressure, gage 80	6.42	6.70	7.00	7.29	7.59	7.86	8.15	8.48	8.75	9.05	9.34	9.62	9.92	10.20	10.50	10.75	11.05	11.36	11.61
Air-pressure, gage 90	6.82	7.14	7.45	7.75	8.05	8.36	8.66	8.98	9.30	9.60	9.94	10.21	10.54	10.85	11.15	11.41	11.74	12.09	12.35
Air-pressure, gage 100	7.25	7.59	7.92	8.25	8.58	8.90	9.22	9.57	9.90	10.20	10.56	10.90	11.20	11.55	11.88	12.16	12.50	12.85	13.15
Air-pressure, gage 110	7.66	8.00	8.36	8.70	9.05	9.40	9.75	10.10	10.45	10.76	11.15	11.50	11.84	12.20	12.52	12.84	13.20	13.55	13.90
Air-pressure, gage 120	8.04	8.40	8.76	9.11	9.50	9.85	10.20	10.58	10.95	11.30	11.66	12.05	12.40	12.76	13.13	13.45	13.82	14.21	14.55

TABLE XV.—STEAM CONSUMPTION OF AIR-COMPRESSORS—(*Continued*)

Two-stage Compression

Pounds steam consumption per indicated horse-power per hour	12	12½	13	13½	14	14½	15	15½	16	16½	17	17½	18	18½	19	19½	20	20½	21	21½
Air-pressure, gage 70	2.82	2.93	3.06	3.17	3.25	3.41	3.52	3.63	3.76	3.87	3.78	4.11	4.23	4.34	4.46	4.58	4.69	4.81	4.93	5.05
Air-pressure, gage 80	3.01	3.13	3.27	3.39	3.51	3.64	3.77	3.89	4.03	4.15	4.26	4.39	4.52	4.64	4.78	4.90	5.02	5.15	5.27	5.41
Air-pressure, gage 90	3.19	3.33	3.46	3.58	3.72	3.86	3.99	4.12	4.26	4.39	4.52	4.66	4.79	4.91	5.06	5.18	5.32	5.46	5.58	5.73
Air-pressure, gage 100	3.37	3.51	3.65	3.79	3.93	4.07	4.21	4.35	4.50	4.63	4.77	4.92	5.05	5.18	5.34	5.47	5.61	5.76	5.90	6.05
Air-pressure, gage 110	3.54	3.70	3.84	3.95	4.14	4.29	4.43	4.57	4.74	4.88	5.02	5.17	5.32	5.46	5.63	5.76	5.91	6.06	6.21	6.37
Air-pressure, gage 120	3.69	3.85	4.00	4.15	4.30	4.46	4.62	4.76	4.93	5.08	5.22	5.39	5.54	5.69	5.86	6.00	6.15	6.32	6.46	6.64
Air-pressure, gage 130	3.83	3.99	4.15	4.30	4.46	4.63	4.79	4.95	5.11	5.26	5.42	5.59	5.75	5.90	6.07	6.22	6.38	6.55	6.70	6.88
Air-pressure, gage 140	3.96	4.13	4.30	4.45	4.62	4.79	4.95	5.11	5.29	5.45	5.60	5.79	5.94	6.10	6.28	6.45	6.60	6.77	6.93	7.11
Air-pressure, gage 150	4.10	4.26	4.45	4.60	4.76	4.95	5.11	5.28	5.46	5.62	5.80	5.98	6.14	6.30	6.49	6.65	6.81	7.00	7.16	7.35

Two-stage Compression—(*Continued*)

Pounds steam consumption per indicated horse-power per hour	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
Air-pressure, gage 70	5.16	5.39	5.63	5.85	6.10	6.33	6.56	6.80	7.04	7.26	7.50	7.74	7.96	8.20	8.45	8.65	8.90	9.15	9.35
Air-pressure, gage 80	5.53	5.77	6.03	6.27	6.53	6.78	7.03	7.27	7.53	7.78	8.03	8.29	8.53	8.78	9.05	9.25	9.53	19.80	10.01
Air-pressure, gage 90	5.85	6.10	6.38	6.65	6.91	7.17	7.44	7.70	7.98	8.24	8.50	8.77	9.03	9.31	9.57	9.81	10.05	10.35	10.60
Air-pressure, gage 100	6.19	6.45	6.74	7.01	7.30	7.57	7.85	8.15	8.42	8.70	8.99	9.26	9.55	9.85	10.10	10.35	10.63	10.95	11.20
Air-pressure, gage 110	6.51	6.80	7.10	7.40	7.70	7.96	8.27	8.58	8.86	9.16	9.46	9.75	10.05	10.35	10.64	10.90	11.20	11.51	11.80
Air-pressure, gage 120	6.78	7.08	7.38	7.69	8.00	8.31	8.61	8.92	9.24	9.53	9.85	10.15	10.46	10.75	11.05	11.35	11.65	12.00	12.27
Air-pressure, gage 130	7.03	7.34	7.66	7.97	8.30	8.61	8.92	9.25	9.57	9.90	10.20	10.52	10.84	11.15	11.48	11.76	12.10	12.42	12.72
Air-pressure, gage 140	7.26	7.59	7.92	8.25	8.60	8.90	9.23	9.56	9.90	10.20	10.56	10.90	11.21	11.55	11.88	12.18	12.50	12.85	13.15
Air-pressure, gage 150	7.50	7.84	8.17	8.51	8.86	9.20	9.55	9.90	10.20	10.55	10.90	11.25	11.60	11.91	12.26	12.58	12.90	13.28	13.60

Mr. H. V. Conrad, who first gave publicity to these tables, remarked as follows:

"The accuracy of these tables in practice depends upon the correct assumption of the indicated horse-power steam consumption. Where this cannot be exactly determined the tables can at best be considered as only an approximation. The tables will, however, be found very useful for quickly making comparisons as to the amount of fuel consumed by the various types of air-compressors, thus showing approximately the expected yearly saving by the use of, for instance, a compound as compared with a simple machine. For example, a straight-line compressor with a steam consumption of 30 lb., single-stage compression to 100-lb. gage, requires 9.9 lb. of steam per 100 cu. ft. of free air compressed. A compressor with duplex steam cylinders, at the same steam consumption, but with compound or two-stage air-cylinders, requires 8.42 lb. of steam. A compressor with compound steam cylinders, non-condensing, with a 26-lb. steam rating and compound air-cylinders, requires 7.3 lb. of steam, while a high-class Corliss compressor using steam at high pressure, with compound steam-cylinders running, condensing with a water rate of 17 lb. (including the condenser) and with compound air-cylinders, requires 4.77 lb. of steam, or one-half as much as in the first example.

"The average man, however, thinks in pounds of coal rather than in pounds of steam. For the purpose of comparison it will usually be better, therefore, to reduce deductions to terms of pounds of coal burned per hour or per day by dividing the steam consumption by 7, since a fair evaporation for average conditions is 7 lb. of water per pound of coal burned. This states the case upon a dollars and cents basis when the price of coal is known."

An inspection of the tables will show that two-stage air compression as compared with single-stage is credited, other things being equal, with a saving of 15 per cent.

Referring to the diagram, Fig. 46, the steam consumption for single-stage compression is represented by the oblique lines which rise from the lower left-hand side, and the lines for two-stage compression start from the lower right hand side. The numbering at the bottom of the table is therefore repeated in reverse order for convenience of reading from either end. For general purposes the diagram may be found more convenient than, and practically as reliable as, the tables.

CHAPTER XIV

POWER FROM COMPRESSED AIR

When compressed air is used just as steam is used in the cylinder of a reciprocating engine, it is quite natural to be comparing the action and value of the two fluids for power purposes, although otherwise this book has little to do with steam. In general it may be said that no special engine or meter is required when air is to be used. Any steam-engine will do, and it may be said that the engine may be expected to operate in a more lively and frictionless way with air than with steam on account of the better lubrication. When air was substituted for steam for driving the main hoisting engines at the mines of the Anaconda Copper Company, Butte, Montana, larger cylinders were placed on the engines, but this was because of the change in the initial working pressure. Where steam at 150 lb. had been used as more economical the air was to be used at 90 lb., as that pressure was better adapted to the other uses of the air in the mine.

Assuming the air to be used expansively as in the steam-engines, the most advantageous point of cut-off being selected and the load being adjusted according to the power available, we find that a cubic foot of air at any given pressure is not worth as much in power developed as a cubic foot of steam at the same pressure, and the diagram, Fig. 47, shows how this can be so.

Here we have one volume of steam and the same of air, both at 100 lb. and each successively expanded, while doing its work, into several additional volumes and until the pressure of each falls below that of the normal atmosphere. It is readily seen that the two expansion lines are quite different, and that the mean effective pressure of the steam is decidedly higher than that of the air.

Thus one volume of steam at 100 lb., represented by the length of the line A–1, reaches atmospheric pressure after expansion to about six and three-quarter times the original volume, while the same volume of air at the same initial pressure drops to the same pressure after expanding to a little over four times its original volume. The mean effective pressure for the steam, taking the entire extent of the diagram, is 27.38 lb., while the M.E.P. for air under the same conditions is 19.51 lb., or only 71 per cent. of the former. As with this cutoff the terminal pressures are below the atmosphere, the entire mean effective pressures are not properly "effective" or available or comparable. At 1/4 cut-off the M.E.P. for steam is 51.93, and for air it is 44.19, or 85 per cent., which looks a little better for the air, but

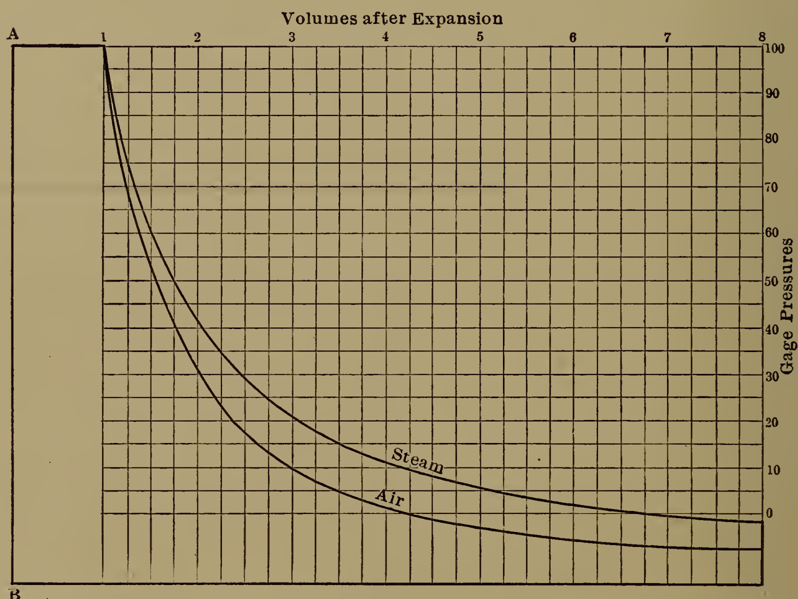

FIG. 47.—Comparative Power Values of Equal Volumes of Steam and of Air.

in this case the terminal pressure of the steam is 11-lb. gage, and some of its power is lost through the exhaust.

This diagram is equally applicable for any other initial pressure below 100, by taking as the measure of volume the length of a horizontal line drawn from the line *AB* to the expansion-line at the given pressure, and taking each repetition of this length horizontally as representing an additional volume. Thus at 60 lb. pressure 1 volume of steam is represented by 1 1/2, and 2 volumes would be represented by 3, and at the intersection of the vertical line marked 3 we find that the steam pressure has fallen to 21 lbs., which is nearly correct. One volume of air at 60 lb. is represented by about 1 3/8 of the diagram-spacing, and 2 volumes would consequently be 2 3/4 of the spaces, and here we find the air pressure to be 13+, which is the correct terminal pressure for air at 60-lb. cut-off at 1/2 stroke, or expanded to double the volume. We may take any section of this diagram as representing, theoretically, an indicator-card either for steam or air, but we cannot take both the steam- and the air-cards and compare them by placing one upon the other, because the lengths of the two cards will not coincide.

Fig. 48 is a theoretical diagram, scale 40, showing both steam

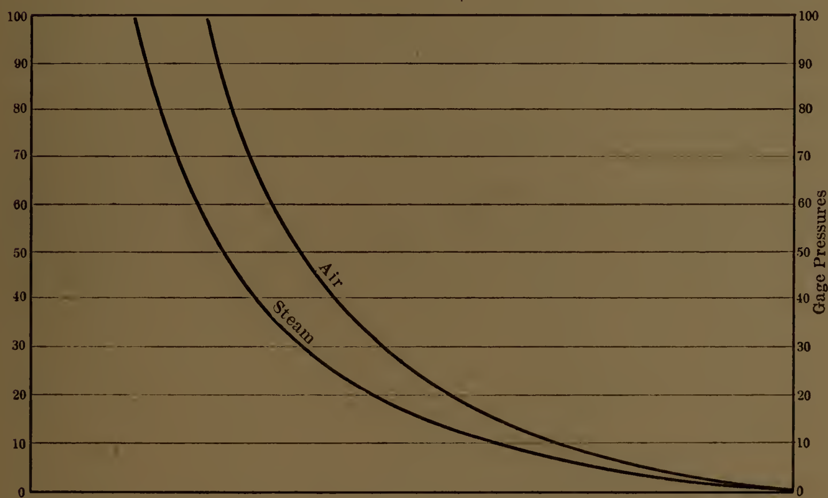

Fig. 48.—Different Expansion Lines of Steam and of Air.

and air expanded to atmospheric pressure at the end of the stroke. In this case the air-line is outside of and above the steam-line, and, of course, represents a higher mean effective pressure, but it is at the expense of a much larger initial volume. The M.E.P. for air filling a cylinder at an initial pressure of 100 lb. for a sufficient portion of the stroke and then expanding (without loss or gain of heat) so that it reaches atmospheric pressure at the end of the stroke will be 41.6 lb. The M.E.P. for steam under the same conditions will be 32.46. The volume of air used will be .2353, while the volume of steam will be 0.1471. If the air gave the same M.E.P. in proportion to its volume, it would be .1471 : 2353 : : 32.46 : 51.9, instead of 41.6, and the greater comparative efficiency of steam under the conditions is 41.6 : 51.9 : : 1 : 1.247, or nearly 25 per cent.

As the expansion of the air here exhibited is adiabatic, its temperature, at least for the latter portion of the expansion, would be below that of the cylinder containing it, and the air would be heated and expanded, rather than cooled, by its surroundings; so that there need be no apprehension that the expansion-line would be below the theoretical, or that there might be still some lurking losses to arise and confront us. The essential difference in an engine or motor to be driven by compressed air instead of steam is a later cut-off for the same initial pressure. This later cut-off develops the paradox that although air has less available power than steam, volume for volume, the same cylinder with the same pressure will develop more power with air than with steam, both being used at the point of highest efficiency or exhausting at a pressure but little above that of the atmosphere.

Table XVI shows the mean effective and terminal pressures for both steam and air at various points of cut-off and for different gage pressures from 50 to 100. Gage pressures are given throughout except when below atmosphere when the absolute pressures are given in heavy face. It is thought that in this way the table will be more serviceable to the general mechanic than if the absolute pressures were given throughout. Nothing is said of the initial temperature of the air, as that would not affect the rate of expansion or the mean effective pressure, after the air entered the cylinder. It should not be forgotten, however, that re-heating the air just before entering the cylinder would increase its volume, and then only a portion of the unit volume assumed would be required to do the same work, and in this way the disadvantage as compared with steam, spoken of above, might be reversed.

A single example may be given to suggest one of the many practical ways of using this table. Say that we take 1 cu. ft. of air at a pressure of 100 lb., cutting off at 1/4 stroke, to get an idea what actual power may be realized from it. This might be in a cylinder with a piston area of 1 sq. ft., then the first foot of piston travel would take the cubic foot of air to fill the space, and a total travel of 4 ft. would allow the air to expand down very close to atmosphere, the M.E.P. for the entire stroke being 44.19, the foot-pounds of work would be: 144 sq. in. piston area ×44.19 M.E.P.×4 ft. piston travel = 25453.44. If the air was used at full pressure, 100 lb., and without cut-off, it would take 4 cu. ft. of air to drive the piston the entire length of the cylinder and the foot-pounds of work would be 144×100×4 = 57,600, or 14,400 ft.-lb. per 1 cu. ft., then 14,400:25,453::1:1.76, which means that the air used expansively in this case would do 76 per cent. more work than the same volume of air at the same pressure without expansion.

To ascertain the actual air consumption in any case computed by the aid of the table at least 25 per cent. should be added to the result obtained, this being not too much to allow for clearance, leakage and friction.

TABLE XVI MEAN EFFECTIVE AND TERMINAL PRESSURES OF STEAM AND AIR AT VARIOUS POINTS OF CUT-OFF AND FOR DIFFERENT GAGE-PRESSURES FROM 50 TO 100 LB.

All pressures given in the table are gage pressures, except where they fall below atmosphere, when the absolute pressures are given and printed in full face.

Initial pressure 50 lb.				
Point of cut-off	Mean steam pressure	Mean air pressure	Terminal steam pressure	Terminal air pressure
.05	**12.12**	**8.87**	**2.69**	**.95**
$\frac{1}{16}$	**14.39**	**10.8**	**3.41**	**1.31**
.10	5.44	1.2	**5.63**	**2.54**
$\frac{1}{8}$	8.95	4.51	**7.13**	**3.47**
.15	10.18	7.62	**8.65**	**4.49**
$\frac{3}{16}$	16.55	11.96	**10.97**	**6.14**
.20	17.9	13.84	**11.75**	**6.74**
.25	22.83	18.45	**14.9**	**9.23**
.30	27.11	23.05	3.08	**11.93**
$\frac{1}{3}$	29.66	25.84	5.22	**13.83**
.35	30.86	27.17	6.3	**14.82**
$\frac{3}{8}$	32.56	29.07	7.92	1.34
.40	34.15	30.87	9.55	2.88
.45	37.03	34.18	12.84	4.11
.50	39.54	37.12	16.12	7.49
.60	43.61	41.98	22.77	16.66
$\frac{5}{8}$	44.44	42.99	24.44	18.53
$\frac{2}{3}$	45.67	44.52	27.24	21.73
.70	46.54	45.6	29.49	24.33
.75	47.64	46.98	32.88	28.34
.80	48.52	48.08	36.27	32.47
$\frac{7}{8}$	49.43	49.26	41.4	38.85
.90	49.64	49.53	43.11	41.03

Initial pressure 60 lb.				
Point of cut-off	Mean steam pressure	Mean air pressure	Terminal steam pressure	Terminal air pressure
.05	13.99	10.23	3.1	1.1
$\frac{1}{16}$	1.61	12.46	3.93	1.51
.10	8.58	3.69	6.49	2.93
$\frac{1}{8}$	12.64	7.51	8.22	4.01
.15	16.37	11.1	9.99	5.21
$\frac{3}{16}$	21.41	16.11	12.66	7.08
.20	22.96	17.7	13.56	7.77
.25	28.75	23.6	2.19	10.65
.30	33.59	28.9	5.87	13.77
$\frac{1}{3}$	36.54	32.13	8.34	.96
.35	37.92	33.66	9.58	2.33
$\frac{3}{8}$	39.87	35.85	11.8	3.85
.40	41.71	37.93	13.22	5.64
.45	45.03	41.75	17.1	10.71
.50	47.94	45.14	20.91	13.26
.60	52.62	50.75	28.59	21.53
$\frac{5}{8}$	53.58	51.92	30.51	23.69
$\frac{2}{3}$	55.01	53.67	33.74	27.94
.70	56.01	54.93	36.34	30.39
.75	57.28	56.52	40.24	35.01
.80	58.29	57.79	44.06	39.78
$\frac{7}{8}$	59.34	59.15	50.07	47.14
.90	59.58	59.46	52.05	49.65

This chapter and that which precedes it have had to do with the cost involved in the use of compressed air for power purposes. The status of compressed air is not based exclusively or even chiefly upon considerations of power cost. It is to be remembered that, while compressed air carries with it the tradition of high cost, it nevertheless was rapidly extending its field of employment when the cost of the air compressed and delivered was two or three times as great as at present, and it is no wonder that it is still finding increased appreciation and still is extending its field of employment.

Especially does air challenge comparison with steam for the operating of the scattered and intermittently operated units of a contractor's plant covering a considerable area of operation. It is concededly impossible to generate steam at a central station and to transmit it by piping to the several machines to be operated, as we do transmit the air, on account of the constant

Initial pressure 70 lb.

Point of cut-off	Mean steam pressure	Mean air pressure	Terminal steam pressure	Terminal air pressure
.05	1.06	11.6	3.52	1.23
$\frac{1}{16}$	3.82	14.12	4.46	1.71
.10	11.73	6.19	7.36	3.32
$\frac{1}{8}$	16.33	10.51	9.32	4.54
.15	20.55	14.58	11.32	5.88
$\frac{3}{16}$	26.26	20.25	14.37	8.03
.20	28.02	22.06	0.37	8.81
.25	34.47	28.74	4.49	12.07
.30	40.07	34.75	6.65	0.6
$\frac{1}{3}$	43.41	38.41	11.45	3.09
.35	44.97	40.15	12.86	4.38
$\frac{3}{8}$	47.19	42.63	14.98	6.36
.40	49.27	44.99	17.1	8.39
.45	53.04	49.31	21.38	12.61
.50	56.33	53.16	25.69	17.0
.60	61.64	59.51	34.4	26.4
$\frac{5}{8}$	62.73	60.84	36.58	28.85
$\frac{2}{3}$	64.34	62.83	40.24	33.03
.70	65.48	64.25	43.19	36.44
.75	66.92	66.05	47.61	41.68
.80	68.07	67.5	52.05	47.08
$\frac{7}{8}$	69.26	69.03	58.75	55.43
.90	69.53	69.38	60.99	58.27

heat radiation of the line and the consequent losses by condensation, besides the trouble caused by expansion and contraction of piping, water hammer, etc., the necessity of providing arrangements for trapping or disposing of the water of condensation, and, in spite of all precautions, the frequent stoppages for repairs entailed. Each steam operated machine, therefore, must have its own boiler and all appurtenances, its own supply of fuel and water. Such isolated and intermittently operated machines, taking the day through, cost in coal actually consumed at least 30 pounds and often much more per horse-power hour of work actually done, or about twice as much as the coal cost of the air operated machines.

With the air driven machine, when the air is piped to it, that ends it, and the operator has only to manipulate the throttle and attend to the lubrication. With the steam driven machine there is not only the cost of the coal actually consumed, but there

Initial Pressure 80 lb.				
Point of cut-off	Mean steam pressure	Mean air pressure	Terminal steam pressure	Terminal air pressure
.05	2.72	12.96	3.93	1.39
$\frac{1}{16}$	6.04	0.78	4.98	1.92
.10	14.87	8.68	8.22	3.71
$\frac{1}{8}$	20.01	13.51	10.42	5.08
.15	24.73	18.06	12.65	6.57
$\frac{3}{16}$	31.12	24.4	1.04	8.97
.20	33.08	26.6	2.18	9.85
.25	40.29	33.89	6.78	13.49
.30	46.55	40.61	11.43	2.44
$\frac{1}{3}$	50.28	44.69	14.56	5.22
.35	52.03	46.64	16.14	6.66
$\frac{3}{8}$	54.51	49.41	18.5	7.88
.40	56.83	52.05	20.88	11.14
.45	61.04	56.9	25.66	15.86
.50	64.72	61.18	30.48	20.81
.60	70.76	68.28	40.21	31.27
$\frac{5}{8}$	71.87	69.76	42.65	34.01
$\frac{2}{3}$	73.68	71.99	46.74	38.68
.70	74.95	73.57	50.03	42.49
.75	76.56	75.59	54.97	48.35
.80	77.84	77.2	59.94	54.38
$\frac{7}{8}$	79.17	78.92	67.43	63.81
.90	79.47	79.31	69.93	66.89

is also the bringing of the coal to the machine, the supplying of the water, the firing and caring for the boiler, with all which that implies, so that there is for each machine the labor of a man or at least the equivalent of one man's labor to be added to the cost of operating.

The equivalent in coal cost of a man's labor is worth considering. Say that coal costs at the machine $4 per short ton. Then if the man's wage is $2 per day that will be 1000 pounds of coal, or 100 pounds per hour, and for 10 horse-power, which is a big allowance for a hoisting engine, this would be 10 additional pounds of coal cost per horse-power hour.

So far, then, as the actual cost of the power used is concerned there is evidently a great saving in the employment of air instead of steam, and on this account alone it is no wonder that the knowing ones choose the air transmission even when there are

Initial Pressure 90 lb.				
Point of cut-off	Mean steam pressure	Mean air pressure	Terminal steam pressure	Terminal air pressure
.05	4.59	14.33	4.34	1.54
$\frac{1}{16}$	8.25	2.95	5.51	2.12
.10	18.02	11.17	9.09	4.1
$\frac{1}{8}$	23.7	16.52	11.51	5.61
.15	28.92	21.55	13.98	7.26
$\frac{3}{16}$	35.97	28.55	2.73	9.92
.20	38.15	30.78	3.99	10.88
.25	46.11	39.04	9.07	14.91
.30	53.02	46.46	14.22	4.27
$\frac{1}{3}$	57.17	50.98	17.67	7.35
.35	59.08	53.13	19.42	8.95
$\frac{3}{8}$	61.82	56.2	22.03	11.39
.40	64.4	59.11	24.65	13.88
.45	69.05	64.45	29.95	19.11
.50	73.11	69.19	33.27	24.56
.60	79.67	77.05	46.02	36.14
$\frac{5}{8}$	81.02	78.69	48.72	39.16
$\frac{2}{3}$	83.01	81.14	53.23	44.33
.70	84.42	82.9	56.88	48.54
.75	86.19	85.12	62.34	55.02
.80	87.61	86.91	67.83	61.69
$\frac{7}{8}$	89.08	88.81	76.1	72.0
.90	89.42	89.24	78.88	75.52

no special conditions as in mining, tunneling, subaqueous work, etc., compelling them to do so.

In addition to the saving in coal cost there are other advantages which air carries with it. In the use of steam there is the time taken to fire up and get the pressure before work commences, there is the warming up process and the working of the water out of the pipes and cylinders every time the machine is started up after standing, none of which delays occur with the air, so that, in constant readiness and instant realization of power to the utmost limit required, the air will every day put in from 10 to 25 per cent. more actual work per day. Stuffing boxes will give no trouble, water will not knock out cylinder heads, pipe joints will not be giving out, there will be no chance of low water in the boiler, no burning of flues or crownsheet, no possible blow up. The cost of repairs, and maintenance will be much less and the certainty of continuous readiness for work will be much

Initial pressure 100 lb.				
Point of cut-off	Mean steam pressure	Mean air pressure	Terminal steam pressure	Terminal air pressure
.05	6.45	.69	4.76	1.69
$\frac{1}{16}$	10.24	4.11	6.03	2.32
.10	21.16	13.66	9.95	4.49
$\frac{1}{8}$	27.38	19.51	12.61	6.15
.15	33.1	25.03	0.31	7.95
$\frac{3}{16}$	40.83	32.71	4.42	10.89
.20	43.21	35.14	5.79	11.92
.25	51.93	44.19	11.36	1.33
.30	59.5	53.32	17.0	6.11
$\frac{1}{3}$	64.02	57.26	20.78	9.48
.35	66.14	59.62	22.69	11.23
$\frac{3}{8}$	69.14	62.98	25.56	13.89
.40	71.96	66.16	28.43	16.64
.45	77.05	72.02	34.23	22.36
.50	81.5	77.21	40.06	28.33
.60	88.69	85.82	51.83	41.01
$\frac{5}{8}$	90.15	87.61	54.79	44.32
$\frac{2}{3}$	92.19	90.32	59.73	49.97
.70	95.89	92.22	63.72	54.59
.75	95.83	94.66	69.7	61.69
.80	97.38	96.61	75.72	68.99
$\frac{7}{8}$	98.99	98.7	84.78	80.28
.90	99.36	99.17	87.82	84.14

greater. While the air driven machines are identical with the steam driven type, the individual boilers and all their appurtenances are dispensed with, the cost of them, as far as it goes, helping to offset the larger costs of the compressed air installation as a whole. The saving in repairs and maintenance with this reduction in the cost of the air operated machines may go to offset the fixed charges entailed in the larger cost of the compressors and piping.

CHAPTER XV

THE AIR-RECEIVER

An air receiver is understood to be quite a necessary adjunct of any air compressing installation, almost regardless of what the air may be used for. It is quite worth while, therefore, to consider what may be assumed to be the functions of the air-receiver, and how completely or otherwise it satisfies the expectations and requirements concerning it. We now have in mind only compressors and their appurtenances which are installed for service sufficiently permanent to warrant the providing of safe and economical working conditions. But this should include practically all air-compressing plants of any considerable capacity, the length of time for which they may be assumed to be installed being nearly always sufficient to justify whatever will make for efficiency.

It is gratifying to note how the great contractors of the day, as for instance some of those having to do with the Catskill aqueduct for the New York water-supply, have learned the economy of all expenditures which guarantee ultimate efficiency, and, having the courage of their convictions, have equipped and operated some of the most perfect air-compressing plants known up to their time.

Compressed air is used as a necessity for operating rock drills when sinking shafts, driving tunnels and in general mining operations, also in sinking caissons for all kinds of subaqueous foundations, etc; but it is also now used as a time and money saver, in still greater volume and for a much wider range of service when the physical conditions do not compel its use. Every extensive manufacturing concern most have its compressed-air supply distributed throughout the works, and for the larger engineering works which are all out of doors, such as dam and waterworks constructions and in the most extensive quarries, compressed air from a central plant is superseding the isolated steam operated pumps, hoists, shovels, stone crushers, concrete mixers, etc.

In all these extensive lines of air service, and in fact in all air-practice, it makes a great difference as to the condition of the air when used, as to the maintenance of constant pressure, suitable temperature, and especially as to the moisture which it may carry, much inconvenience and loss of time resulting when the air is not kept dry and clean.

There is always a pretence made of attending to this necessity, and the air-receiver too often might not untruthfully be said to be the embodiment of this pretence. Who would think for a moment of installing an air-compressor without a good-sized air-receiver as close to it as possible? The man who would set up an air-compressor without an air-receiver would lose all his respectability among engineers. Respectability, or what corresponds to it, has perhaps a good deal to do with the installation of an air-receiver such as it too frequently is to-day than anything else, but whatever the ostensible reason for its existence it never fails to be more or less of a disappointment.

If you were to ask the whys and wherefores of an air-receiver, you would probably be told that "everybody always uses them." If you pressed the matter still further, you would most likely be told that the receiver is needed for the storage of air and the steadying of the flow. Well, the total capacity of the air-receiver usually installed with an air-compressor does not exceed the output of the compressor for one minute. If the compressor is running continuously, and if the air is being used as fast as it is delivered, the apparatus driven by the air must lose its vim instantly and must come to a dead stop within a single minute if the compressor stops. So it can be readily seen that the air-receiver as a storage adjunct is of little value.

Receivers do not Cool the Air.—Builders and sellers can afford to have you pooh-pooh the storage service of air-receivers, but they will still assert the two-fold need of them for cooling the air and for getting rid of the moisture in it, the presence of the latter being the most serious objection to the air in use and the cause of the most trouble. Now as to the cooling of the air, how can the receiver do it? To cool the air as it comes from the compressor, there must be a cooling surface or material for the air to come in contact with, and to which it may impart its heat. The air-receiver, however, is a plain cylindrical shell with no pretence of a cooling device or any arrangement of the sort within it, and the air merely passes through it hurriedly, emerging at the other end practically as hot as when it entered, giving up its heat more or less rapidly to the inner surfaces of the pipes through which it travels, for which cooling effect the pipes and not the receiver should have the credit.

The Receiver does not dry the Air.—While the air-receiver is not, thus, a cooler of the air, it also is not, and for the same reason, an abstractor of the moisture in the air, this being the most desirable service which it could render, and which it has been too readily assumed to do. It is well enough understood that atmospheric air—free air—always carries moisture and also always has capacity, or as we might say appetite for more, up to the point of saturation, when its avidity suddenly ceases. The moisture-carrying capacity of the air rises very rapidly with its rise in temperature, and diminishes, but not so rapidly, with rise of pressure. As the pressure must always be at the highest point just when the air is leaving the compressor, if we can then reduce its temperature to the lowest point, the air will be in a condition to surrender so much of its moisture that none will be found later to cause trouble, when lower pressures and perhaps higher temperatures are reached further along the line. It being thus sufficiently evident that the best place for drying the air is as near the compressor as possible, the first requisite is an efficient cooler, or rather after cooler, for the air.

The air-receiver, as has been said, has never been a cooler of the air, and the builders are now testifying to this fact by the number of aftercoolers they are offering. The best of these are highly efficient, and the air after going direct from the compressor and through the intercoolers is in the precise condition desired —that is, of high pressure and low temperature—for the surrender of its moisture. For the important service which the aftercooler may render, nothing could be expected to work more cheaply. There is only the first cost of the apparatus and connections, and then a sufficient supply—not large—of cool, free-running water.

Now, surely, close to the aftercooler, the air-receiver should be available and effective for abstracting the excess of moisture which the aftercooler has liberated and for passing the air along so dry that there will be no trouble from moisture in the air anywhere along the line after that. Unfortunately the typical air-receiver still persists in its inefficiency, and the air passes through and out of it wet or carrying a considerable amount of condensed but not separated moisture.

It is to be remembered that when the saturation point is lowered by the lowering of the temperature of the air, and there is a relinquishing of the surplus moisture by condensation into water, that water still remains in the air, as mist or fog, and what is then needed is a separator. Any of the separators which are successful in drying steam are equallly efficient in taking the liberated water out of the air, when it is in the condition here spoken of. The efficiency of such separators is due to the habit which water has, and which liquids like alcohol, benzine, etc.,

Fig. 49.—Draining the Water from the Line.

do not possess, of wetting or clinging to the surfaces with which it comes in contact. A constant repetition of this wetting process causes the water to drip or flow off and accumulate in pockets provided, from which it may at intervals be drawn off.

Draining the Pipe Line.—Fig. 49 tells its own story of what to do when there is a low spot in a long pipe line. There was inserted here a short plain cylinder with the line pipe entering it at one end and leaving it at the other end close to the top, the bottom of the cylinder forming a water pocket from which the water was drawn at frequent intervals in considerable quantities, and there was no freezing up of drills on that line.

Improve the Air-receiver.—An air-receiver of the common type, preferably horizontal, if merely provided with a series of baffle-plates (which might be added without much additional cost) would be an efficient separator, provided that the air passing through it were in the prescribed condition of high pressure and low temperature. Indeed it would seem to be quite possible to combine in one apparatus the two functions of cooling the air and separating the freed water from it, and it might also be made large enough to constitute a storage capacity equal to that of the plain air-receiver. It would seem to be almost an absurdity to be still installing the latter alone. We may look for some enterprising manufacturer to be putting on the market a combination air-receiver covering the three separate functions of aftercooler, separator and air holder. Since all these functions must be provided for in good practice, a single construction should be more compact, more efficient and cheaper than three separate devices. The writer has in mind one of the largest compressor installations in which each compressor unit has an aftercooler and then a receiver, with a separator added, after the completion of the installation, as a necessity. The separator should have immediately followed the aftercooler and the receiver might perhaps as well have been dispensed with, as the air was delivered into long lines of large pipe.

The intercooler or the aftercooler should be expected, or more properly required, to cool the air to within 10 degrees of the temperature of the cooling water. In 1906 Mr. H. V. Haight, of Sherbrooke, Canada, had the designing of two air compressors each of 4000 cu. ft. per min. free air capacity, and in the contract it was specified that the temperature of the air passing from the low pressure to the high pressure cylinder should be reduced to within 15 degrees of the temperature of the cooling water, with a penalty of $150 per degree in excess of this and a bonus of $150 for every degree below the 15. In actual running, at the required speed and pressure, the temperature of the air was reduced to within 5 degrees of that of the cooling water, and a bonus of $1500 was earned for the builders.

There are indications that the air-receiver is soon to be doing better things. That it will be enabled to make good in all the particulars in which it is now deficient is too much, perhaps, to expect, but in what has been its principal accredited function, that of air storage, there is a notable recent development.

Compressed-air Storage.—It may be said that compressed air wherever employed is always used more or less intermittently, and never at any constant rate, except in cases where the entire output of a compressor or of an entire compressing plant, is employed in a single water-pumping operation, so that the desirability or the necessity of air storage is not to be ignored. The air-receiver as generally installed not only does not hold enough to be of much account, but it does not maintain a constant pressure for a moment if the intake and the output vary, and so we look to the compressor for help. To insure somewhat reliable maintenance of pressure and volume it is the practice to provide a maximum compressing capacity somewhat in excess of the maximum demand, and then to automatically reduce the speed of the compressor as the rate of consumption diminishes. Even this arrangement usually does not completely satisfy the fluctuating requirements, and so we have various unloading or choking contrivances which will still more reduce the output without actually stopping the machine. However satisfactory the results thus obtained may be, it is evident that they are secured only by more or less complication of apparatus and a sacrifice of the essential conditions of power economy in the running of the machine.

A magnificent opportunity for the solution of this air-power storage problem opened to the engineers of the Anaconda Copper Mining Company, at Butte, Mont., when it was proposed to find a cheaper means of driving their great mine hoists than by the use of steam. There were at Butte 25 large steam-driven hoists with an aggregate capacity of 40,000 h.p., but the service required of the hoists was so intermittent, and the actual time of working of each was so short, that it was estimated that 4000 h.p. in constant operation would be sufficient for all the requirements, but it was imperative that the power should be always ready and always sufficient for each individual hoist.

The cost of steam had been about $80 per horse-power per year, while electric horse-power per year could be had for about $25. There were, however, serious objections to the adoption of the electric drive for each separate hoist, besides the enormous first cost of such an installation. With electric drive, also, there could be no power storage, so that it would be necessary at times to have current available for nearly all the hoists at once.

So far as the steam hoisting engines were concerned, they could be adapted to the using of compressed air at comparatively slight cost, if only the power-storage problem could be solved, so that a constant drive of sufficient average capacity could be made able to take care of the peak loads whenever they should occur, even up to the running of all the hoists at once. The problem has been solved with a success and completeness seldom surpassed in great engineering undertakings.

It is not intended here to give more than the briefest sketch of the compressor installation and operation, the purpose being only to call attention to the air-storage scheme.

The electric current which drives the compressors is transmitted 130 miles from the new plant at the Great Falls Water Power and Townsite Company, at Rainbow falls, just below the Great falls on the Missouri river. There are three compressors, each with a direct-connected Westinghouse motor of 1500 maximum horse-power. The compressors, furnished by the Nordberg Manufacturing Company, are two-stage machines of the highest class, with low-pressure cylinders 50 in. in diameter and high-pressure cylinders 30 in. in diameter, and a common stroke of 48 in. The combined free-air capacity of the three compressors I would estimate roughly at 20,000 cu. ft. per minute (not knowing the builder's specifications as to speed, etc.).

From the compressors the air passes to the battery of air receivers Fig. 50. These are vertical, each 10 ft. in diameter and 30 ft. high, their combined cubical content being, say, 70,000 cu. ft., which, at 90 lb. gage pressure may be said to equal 500,000 cu. ft. of free air, a volume which it would take the combined compressors nearly half an hour to compress and deliver. This is very different, to begin with, from the less than one minute capacity of the air receiver usually provided.

But there is a more important feature and a greater difference and advantage to be noted in the present installation as compared with long-established air-receiver practice. The too familiar experience is that as soon as any air is withdrawn from the receiver in excess of what the compressor is delivering, or if for any reason the compressor stops, the pressure in the receiver falls rapidly and constantly with the drawing of the air. Under the arrangement here being considered, when the quantity of air contained in the receivers is diminished by any air consumption exceeding the delivery, instead of a drop of pressure rendering the remaining contents of the receiver ineffective and useless, the pressure is maintained and the entire contents of the whole battery of receivers, including the original inert filling of air at atmospheric pressure, can be used at full pressure and effectiveness until the receivers are emptied. In practice the withdrawal of the air never goes as far as this. As these compressors run all day and all night, when there is at any time a simultaneous call for operating an unusual number of hoists, there is always the full capacity of the working compressors and also the entire

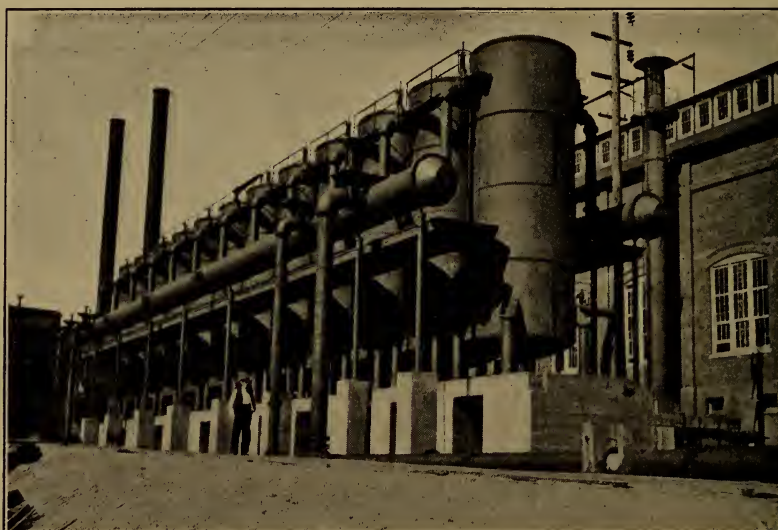

Fig. 50.—Constant Pressure Air Storage—Anaconda Copper Mining Company, Butte, Montana.

contents of the battery of receivers to draw from until the unusual and excessive demand for air ceases. When there is such an unusual simultaneity of hoisting it is necessarily succeeded by a period when the hoisting and the demand for air are less than the compressor output, and then the receivers are automatically filled again.

How the Air-pressure is Maintained.—The device by which the air-pressure is maintained in the receivers notwithstanding the diminution of the contained volume of air is essentially a simple one. It is accomplished by the use of a standpipe or its equivalent, the same as in waterworks service. On a side hill at an elevation sufficient to give the required gage pressure of 90 lb., there is an open water tank 100 ft. in diameter and 15 ft. deep. A depth of 10 ft. in this tank gives a water capacity somewhat greater than the total cubic capacity of the battery of air-receivers. As 2.3 ft. of water gives 1 lb. pressure, the mean elevation of the tank above the receivers should be $90 \times 2.3 = 207$ ft. There is a large pipe connection from the bottom of this tank to a horizontal pipe in free communication with the lower ends of all the receivers. No valves of any kind are required, and little, if anything, need be allowed for the friction of the water in the pipes, it being free to flow in either direction, according to the changes of the volume of air in the receivers. No safety valves are required, and it is impossible to produce any pressure in the receivers greater than that due to the head of water.

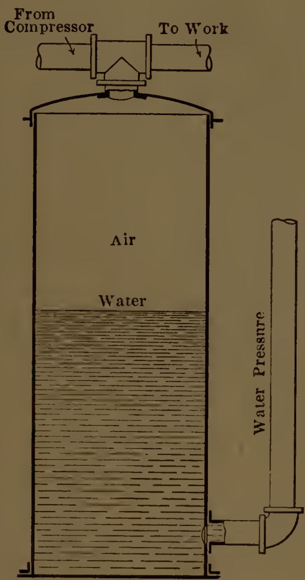

FIG. 51.—Constant Pressure Air Receiver.

The compressed air as it is delivered from the operating compressors does not pass through the receivers, and, indeed, does not enter them at all except when the air production is greater than the consumption at the moment, when the surplus passes into the receivers, driving out and up into the elevated tank some of the water at the bottom of the receivers. When the call for air is greater than the compressor supply then the deficiency is made up by a flow of air from the receivers, the water from the tank displacing it.

The contact of the air with the water does not make it any wetter, as after compression it is quite certain to be saturated with water in any case. In the service for which this air is used, there is no call for "dry" air, as special means are provided for heating the air before it enters the hoisting engines, and moisture would be an advantage rather than otherwise.

Long-continued records of the hoists as they had been run by steam made it possible to compute the compressor capacity required, and to adapt the compressors to the work so that they could be operated at the point of best economy.

The plant is unique as it stands, but in the use of the elevated tank it sets an example which, since there can be no monopolizing of the principle, we may expect will in time be widely adopted. For the maintenance of a constant air-pressure with considerable storage capacity it seems to recognize and fill a long persisting requirement. Fig. 51 may serve as a reminder of the principle of it. The elevations which will give gage pressures of 50, 60, 70, 80, 90 and 100 lb. are, respectively, 115, 138, 161, 184, 207 and 230 ft. These heights can, of course, be secured as well by sinking the receivers as by elevating the tanks, or by a combination of both until the vertical difference is secured.

It is to be noted that no water is consumed or wasted in the operation; it simply flows back and forth out of the receivers and in again as the volume of air in storage increases or diminishes. Why not then dispense with the special elevated water tank and connect direct to a city or other water service where sufficient pressure is maintained, as it usually is? If the water pressure is constant and is somewhat greater or less than the air-pressure desired the latter might be adjusted by placing the air-receiver above or below what would otherwise be its normal position.

CHAPTER XVI

PIPE TRANSMISSION

In computations having to do with the profitable transmission of air from the compressor to the work there are several particulars to be considered, such as the initial pressure of the air, the volume to be transmitted per unit of time, say 1 minute, the distance to be traversed and the drop of pressure that may be permitted, the latter particular depending upon the size of pipe; and computations in air transmission usually are for the purpose of determining the pipe size.

The ultimate decision as to the size of pipe in a given case is often in the nature of a compromise such as prevails in determining most business conditions. The pipe may be so large that there will be no appreciable drop of pressure, but this the cost of piping will prohibit; and then if it is sought to save too much in pipe cost by making the pipe too small there will be a constant loss in effective working pressure which will overbalance the saving. For the adjustment of the compromise between the diameter of the pipe and the speed of the transmission experience is the most reliable arbiter, and in practice limits have been found within which those who work most profitably confine themselves.

In considering the operation of compressing the air we base our computations upon the volume of *free air* taken in and compressed, but in questions relating to the transmission of the air after compression it is necessary to consider the actual volume of the air during the transmission, or usually either at the beginning or at the termination of the transmission. As the air soon attains the temperature of the pipe and its surroundings its temperature need not generally be taken into account as affecting the volume.

The volume of free air transmitted (figuring backward) may be assumed to be directly as the absolute pressure, or as the number of atmospheres to which the air has been compressed. Thus, if we have a known volume of compressed air flowing

TABLE XVII.—VOLUME OF AIR TRANSMITTED IN CUBIC FEET PER MINUTE IN PIPES OF VARIOUS DIAMETERS

Velocity of flow		Diameter of pipe in inches														
Feet per second	Feet per minute	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	8	10	12	16	20	24
1	60	0.327	0.623	0.848	1.31	2.045	2.95	4.00	5.24	8.18	11.78	20.94	32.73	47.12	83.77	130.9	188.5
2	120	0.655	1.246	1.696	2.62	4.09	5.89	8.02	10.47	16.36	23.56	41.89	65.45	94.25	167.5	261.8	377.0
3	180	0.982	1.869	2.544	3.93	6.13	8.84	12.02	15.7	24.5	35.3	62.8	98.2	141.4	251.3	392.7	565.5
4	240	1.31	2.492	3.392	5.24	8.18	11.78	16.03	20.9	32.7	47.1	83.8	131.0	188.0	355.0	523.0	754.0
5	300	1.64	3.11	4.24	6.54	10.22	14.7	20.04	26.2	41.0	59.0	104.0	163.0	235.0	419.0	654.0	942.0
6	360	1.96	3.74	5.09	7.85	12.27	17.7	24.05	31.4	49.1	70.7	125.0	196.0	283.0	502.0	785.0	1131.0
7	420	2.29	4.36	5.93	9.16	14.31	20.6	28.06	36.6	57.2	82.4	146.0	229.0	330.0	586.0	916.0	1319.0
8	480	2.62	4.98	6.79	10.5	16.36	23.5	32.06	41.9	65.4	94.0	167.0	262.0	377.0	670.0	1047.0	1508.0
9	540	2.95	5.61	7.63	11.78	18.4	26.5	36.07	47.0	73.0	106.0	188.0	294.0	424.0	754.0	1178.0	1696.0
10	600	3.27	6.23	8.48	13.1	20.4	29.5	40.09	52.0	82.0	118.0	209.0	327.0	471.0	838.0	1309.0	1885.0
12	720	3.93	7.48	10.18	15.7	24.54	35.3	48.1	63.0	98.0	141.0	251.0	393.0	565.0	1005.0	1571.0	2262.0
14	840	4.58	8.72	11.87	18.3	28.6	41.3	56.1	73.0	114.0	165.0	293.0	458.0	659.0	1172.0	1833.0	2639.0
15	900	4.91	9.35	12.72	19.6	30.7	44.2	60.1	78.0	122.0	177.0	314.0	491.0	707.0	1256.0	1963.0	2827.0
16	960	5.23	9.9	13.56	20.9	32.7	47.2	64.1	84.0	131.0	189.0	335.0	524.0	801.0	1340.0	2094.0	3016.0
18	1080	5.89	11.21	15.26	23.5	36.8	53.1	72.1	94.0	247.0	212.0	377.0	589.0	848.0	1508.0	2356.0	3393.0
20	1200	6.54	12.46	16.9	26.2	40.9	59.0	80.2	105.0	164.0	235.0	419.0	654.0	942.0	1675.0	2618.0	3770.0
22	1320	7.19	13.7	18.65	28.8	45.0	64.9	88.2	115.0	180.0	258.0	461.0	720.0	1036.0	1842.0	2879.0	4147.0
24	1440	7.85	14.95	20.35	31.4	49.1	70.8	96.2	125.0	196.0	283.0	502.0	785.0	1131.0	2010.0	3141.0	4524.0
25	1500	8.18	15.56	21.2	32.7	51.1	73.8	100.2	131.0	204.0	294.0	523.0	818.0	1178.0	2094.0	3272.0	4712.0
26	1560	8.5	16.18	22.05	34.0	53.2	76.7	104.2	136.0	213.0	305.0	544.0	851.0	1225.0	2178.0	3403.0	4900.0
28	1680	9.16	17.44	23.72	36.6	57.2	82.6	112.2	146.0	229.0	330.0	586.0	916.0	1319.0	2346.0	3665.0	5278.0
30	1800	9.8	18.69	25.44	39.3	61.3	88.5	120.3	157.0	245.0	353.0	628.0	982.0	1414.0	2513.0	3927.0	5655.0
32	1920	10.46	19.94	27.14	41.9	65.4	94.4	128.3	168.0	262.0	378.0	670.0	1047.0	1508.0	2680.0	4188.0	6032.0
34	2040	11.11	21.18	28.83	44.5	69.5	100.3	136.3	178.0	278.0	400.0	712.0	1113.0	1602.0	2848.0	4450.0	6409.0
35	2100	11.44	21.80	29.68	45.8	71.5	103.2	140.3	183.0	286.0	411.0	733.0	1145.0	1649.0	2932.0	4581.0	6597.0
36	2160	11.77	22.43	30.53	47.1	73.6	106.2	144.3	188.0	294.0	423.0	754.0	1178.0	1696.0	3015.0	4712.0	6986.0
38	2280	12.42	23.67	32.22	49.8	77.7	112.1	152.3	198.0	311.0	447.0	795.0	1244.0	1791.0	3183.0	4974.0	7163.0
40	2400	13.08	24.92	33.92	52.4	81.8	118.0	160.3	209.0	327.0	471.0	837.0	1309.0	1885.0	3350.0	5236.0	7540.0

through a pipe at a pressure at 75 lb., gage, or say 6 atmospheres, the actual volume of free air will be six times as much.

For computing or comparing cases of air transmission the linear velocity of flow in the pipe is generally adopted and is the more convenient form of statement. Table XVII gives the actual volume per minute of air passing through a pipe of given diameter when the linear velocity of flow is known. This is merely a convertible table of pipe capacities and will be useful as such in determining the size of pipe required for a given service.

It is generally considered that for economical transmission the actual velocity in main pipes should not exceed 20 ft. per second, or 1200 ft. per minute, the latter quantity being more convienient as the minute is the time unit which generally prevails in compressed-air practice.

It would be well if more attention were given to the capacities of the branch or distributing pipes employed. Why should there ever be higher velocities of flow here than in the mains? In actual practice it often occurs that, while the main pipe is large enough for the transmission, the smaller pipes or those through which the air is finally carried to the individual machines are too small, and velocities as high as 3000 ft. per minute or more are not infrequent.

Even when the aggregate sectional area of the branch pipes is equal to that of the main which supplies them, so that the velocity of the flow should be the same as in the main, it is found in practice that to maintain that flow in the smaller pipes entails a more rapid drop of pressure, this drop of pressure increasing the volume and thus still further retarding the flow.

The inadequacy of the small pipes of equal aggregate sectional area with the large pipe is due chiefly to the greater pipe surface and the additional friction caused by it. The interior surface of four pipes, 1 in. in diameter, for instance, is equal to that of one pipe 4 in. in diameter, but to have a pipe area equal to that of the 4-in. pipe we must have 16 pipes 1 in. in diameter, with consequently four times as much pipe surface for the air to rub against, as we might say.

It is found that the diameters of branch pipes to equal that of the mains which supply them, or *vice versa*, should be as the square root of the fifth power of the several diameters, and upon this basis Table XVIII has been computed. The diameters of the

TABLE XVIII.—RELATIVE CAPACITIES OF AIR MAINS AND BRANCHES

Diameters of mains	Diameters of branches																											
	1		1½		2		2½		3		3½		4		5		6		8		10		12		16		20	
	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b	a	b
1½	2.7	.361																										
2	5.6	.178	2.1	.476																								
2½	9.8	.102	3.6	.2777	1.7	.588																						
3	15.5	.0645	5.6	.1785	2.7	.361	1.6	.625																				
3½	23	.0434	8.3	.1205	4.	.25	2.3	.434	1.4	.714																		
4	32	.0312	11.6	.0862	5.6	.1785	3.2	.3125	2.1	.476	1.3	.769																
5	56	.0178	20.3	.0492	9.8	.102	5.6	.1785	3.6	.2777	2.4	.416	1.7	.588														
6	88	.0113	32	.0312	15.5	.0645	8.9	.1123	5.6	.1785	3.8	.263	2.7	.361	1.6	.625												
8	181	.0055	66	.0151	32	.0312	18	.0535	11.6	.0862	7.8	.128	5.6	.1785	3.2	.3125	2.1	.476										
10	316	.0031	115	.0087	56	.0178	32	.0312	20	.05	13.8	.0725	9.8	.102	5.6	.1785	3.6	.2777	1.7	.558								
12	498	.002	181	.0055	88	.0113	50	.02	32	.0312	21	.0476	15.5	.0645	8.9	.1123	5.6	.1785	2.7	.361	1.6	.625						
16	....	.....	373	.0027	181	.0055	103	.0097	66	.0151	44	.0227	32	.0312	18	.0555	11.6	.0862	5.6	.1785	3.2	.3125	2.1	.476				
20	....	.....	652	.0015	316	.0031	181	.0055	115	.0087	78	.0128	56	.0178	32	.0312	20	.05	9.8	.102	5.6	.1785	3.6	.2777	1.7	.588		
24	....	.....	....	.....	498	.002	286	.0034	181	.0055	123	.0081	88	.0118	50	.02	32	.0312	15.5	.0645	8.9	.1123	5.6	.1785	2.7	.361	1.6	.625

larger pipes or mains are given in the first vertical column at the left, and the numbers of small or branch pipes required to equal the carrying capacities of the mains are given in the succeeding columns under the different diameters. The column *a* under each branch-pipe size gives the number of the smaller pipes required, and in column *b* is the reciprocal of this number, which will often be convenient as a multiplier, it representing also the fraction of main pipe equivalent to one of the branch pipes of the given diameter.

Referring to the table, we note that 32 pipes 1 in. in diameter are required to equal the carrying capacity of a 4-in. pipe, which is just double the ratio of the squares of the diameters 16 : 1. So also it takes 316 pipes 1 in. in diameter to equal a 10-in. pipe, or more than three times the ratio of the squares of the diameters.

The popular impression has been that great power losses are unavoidable in the transmission of air through pipes except for very short distances, but the facts do not sustain this view. We cannot do or get anything for nothing and, of course, there is some diminution of available power in the transmission of compressed air, but unless the piping system is very bad and inadequate the transmission losses are among the lesser ones which have to be considered, and not at all to be compared with the loss of power through the heating of the air in compression or through the loss of volume by the cooling of re-expansion when the air is finally put to use. It would perhaps be quite as truthful, and more in accord with business practice in other lines, if those things were thought of and spoken of as expenses rather than as losses.

In line with the transmission of compressed air, and subject to the same laws, is the transmission of natural gas from the wells for long distances to towns and cities where it can be most profitably used. Artificial gas also, used so largely for illuminating, for heating and other industrial purposes, as spoken of elsewhere, is coming to be more and more conveyed at high and higher pressures, so that here also the transmission question is of importance.

There are no formulas for air or gas transmission which can be said to be absolutely right, or which are to be implicitly relied upon regardless of all conditions. Formulas which may suffice for ordinary practice, and which within reasonable limits tell us what may be expected to be the loss of pressure for a flow reaching

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a certain limited distance, would, if the flow was continued far enough, show a gain of pressure instead of a loss.

Table XIX of loss of head during the flow of compressed air through line pipes was computed by what may be called the Rix-Johnson formula. It gives results which are approximately correct in practice, and I have reason to believe that it is now used in the United States more generally than any other in practical compressed-air work. J. E. Johnson, Jr., gave Church's familiar formula in a simplified form in the *American Machinist,* July 27, 1899, which was, in substance, as follows:

$$R=\frac{KV^2L}{D^5}$$

where

V = Volume of free air in cubic feet per minute;
L = Length of pipe in feet;
D = Diameter of pipe in inches;
K = A numerical constant, which Johnson fixed as 0.0006;
R = Difference between the squares of the initial and terminal absolute pressures in pounds per square inch; that is, $p^2{}_1-p^2{}_2$.

E. A. Rix, of San Francisco, uses Johnson's formula with 0.0005 for the value of K. In my own practice I have found it more convenient to use the reciprocal of this value of K, that is, 2000, and transfer it to the divisor, where in actual numerical operations it almost invariably disappears at once by cancellation. I also find it more convenient for my pencil habit to use small letters instead of capitals; hence my working formula takes the form,

$$p^2{}_1-p^2{}_2=\frac{v^2\,l}{2000\,d^5}$$

This, of course, may be transformed to suit the special requirements, as:

$$l=\frac{2000\,d^5\,(p^2{}_1-p^2{}_2)}{v^2}$$

$$v=\sqrt{\frac{2000\,d^5\,(p^2{}_1-p^2{}_2)}{l}}$$

$$d^5=\frac{v^2 l}{2000\,(p^2{}_1-p^2{}_2)}$$

TABLE XIX.—LOSS OF HEAD IN TRANSMISSION THROUGH PIPES

1-in. pipe

Initial pressure, 75 lb., gage						Initial pressure, 90 lb., gage						Initial pressure, 105 lb., gage					
Free air per minute, cubic feet	Flow in pipe per minute, feet	Length of pipe in feet				Free air per minute, cubic feet	Flow in pipe per minute, feet	Length of pipe in feet				Free air per minute, cubic feet	Flow in pipe per minute, feet	Length of pipe in feet			
		100	200	300	400			100	200	300	400			100	200	300	400
12.5	385	0.05	0.09	0.13	0.17	12.5	330	0.04	0.08	0.11	0.15	12.5	289	0.03	0.06	0.10	0.13
25.0	771	0.18	0.35	0.53	0.70	25.0	661	0.15	0.30	0.45	0.59	25.0	578	0.12	0.25	0.40	0.52
50.0	1543	0.69	1.40	2.11	2.83	50.0	1322	0.59	1.19	1.81	2.41	50.0	1157	0.52	1.05	1.56	2.10
75.0	2314	1.58	3.18	4.82	6.49	75.0	1984	1.35	2.72	4.10	5.51	75.0	1736	1.10	2.25	3.40	4.68
100.0	3086	2.82	5.74	8.76	11.89	100.0	2645	2.41	4.88	7.41	10.00	100.0	2313	2.11	4.25	6.42	8.65

$1\frac{1}{2}$-in. pipe

		Length of pipe in feet						Length of pipe in feet						Length of pipe in feet			
		250	500	750	1000			250	500	750	1000			250	500	750	1000
25	339	0.06	0.12	0.18	0.23	25	291	0.05	0.10	0.15	0.20	25	254	0.04	0.08	0.12	0.17
50	679	0.23	0.46	0.69	0.92	50	582	0.19	0.30	0.59	0.79	50	510	0.17	0.33	0.50	0.68
100	1358	0.92	1.85	2.79	3.74	100	1164	0.79	1.58	2.38	3.18	100	1019	0.69	1.38	2.07	2.76
150	2037	2.08	4.22	6.39	8.64	150	1746	1.78	3.59	5.43	7.30	150	1528	1.53	3.14	4.65	6.32
200	2716	3.74	7.64	11.73	16.06	200	2328	3.18	6.47	9.87	13.39	200	2037	2.76	5.55	8.51	11.54

2-in. pipe

		Length of pipe in feet						Length of pipe in feet						Length of pipe in feet			
		200	500	1000	2000			200	500	1000	2000			200	500	1000	2000
100	764	0.17	0.43	0.87	1.75	100	655	0.15	0.37	0.75	1.50	100	573	0.13	0.32	0.65	1.30
150	1146	0.39	0.98	1.97	3.99	150	982	0.34	0.83	1.69	3.40	150	859	0.29	0.74	1.47	2.95
200	1528	0.69	1.75	3.54	7.24	200	1310	0.59	1.51	3.02	6.48	200	1146	0.53	1.32	2.64	5.28
250	1910	1.09	2.31	5.60	11.60	250	1637	0.93	1.97	4.76	9.75	250	1433	0.83	2.09	4.17	8.35
300	2292	1.57	4.92	8.18	17.28	300	1965	1.35	3.45	6.92	14.37	300	1719	1.22	3.06	6.13	12.25

2 1/2-in. pipe

		Length of pipe in feet						Length of pipe in feet						Length of pipe in feet			
		1000	1500	2000	2500			1000	1500	2000	2500			1000	1500	2000	2500
100	489	0.29	0.44	0.59	0.73	100	419	0.25	0.38	0.51	0.62	100	367	0.22	0.32	0.43	0.55
200	978	1.16	1.74	2.32	2.89	200	838	0.98	1.47	1.96	2.47	200	734	0.86	1.29	1.73	2.15
300	1467	2.6	3.91	5.25	6.64	300	1257	2.22	3.34	4.45	5.64	300	1100	1.94	2.88	3.87	4.95
400	1956	4.64	6.96	9.30	12.20	400	1676	3.98	6.00	8.00	10.25	400	1467	3.45	5.18	7.00	8.86
500	2450	7.50	10.00	15.10	18.90	500	2100	6.25	9.38	12.56	15.72	500	1838	5.45	8.20	11.15	14.17

3-in. pipe

		Length of pipe in feet						Length of pipe in feet						Length of pipe in feet			
		1000	1500	2000	2500			1000	1500	2000	2500			1000	1500	2000	2500
100	339	0.12	0.17	0.23	0.29	100	291	0.10	0.15	0.20	0.25	100	254	0.09	0.13	0.17	0.22
200	679	0.46	0.69	0.92	1.15	200	582	0.39	0.59	0.79	0.99	200	509	0.35	0.53	0.69	0.87
400	1358	1.86	2.79	3.75	4.69	400	1164	1.58	2.39	3.19	3.99	400	1019	1.41	2.12	2.76	3.48
600	2037	4.21	6.39	8.65	10.81	600	1746	3.59	5.43	7.31	9.14	600	1528	3.13	4.83	6.48	8.16
800	2716	7.64	11.74	16.11	20.15	800	2328	6.47	9.87	13.43	16.79	800	2037	5.74	8.72	11.78	14.65

4-in. pipe

		Length of pipe in feet						Length of pipe in feet						Length of pipe in feet			
		1000	1500	2000	2500			1000	1500	2000	2500			1000	1500	2000	2500
250	477	0.17	0.26	0.34	0.43	250	409	0.15	0.24	0.29	0.37	250	358	0.12	0.19	0.26	0.32
500	955	0.68	1.02	1.37	1.71	500	818	0.58	0.82	1.17	1.47	500	716	0.51	0.76	1.02	1.28
750	1432	1.54	2.32	3.10	3.89	750	1227	1.31	1.98	2.65	3.32	750	1074	1.15	1.73	2.31	2.89
1000	1910	2.76	4.17	5.60	7.06	1000	1637	2.35	3.55	4.76	5.98	1000	1433	2.05	3.09	4.14	5.20
1250	2387	4.34	6.52	8.91	11.29	1250	2046	3.49	5.59	7.53	9.51	1250	1790	3.22	4.86	6.53	8.22

6-in. pipe

		Length of pipe in feet.						Length of pipe in feet						Length of pipe in feet			
		1000	2500	5000	10,000			1000	2500	5000	10,000			1000	2500	5000	10,000
1000	849	0.36	0.89	1.81	3.65	1000	727	0.31	0.77	1.54	3.10	1000	637	0.27	0.67	1.34	2.70
1500	1273	0.81	2.04	4.12	8.46	1500	1091	0.63	1.74	3.51	7.15	1500	955	0.61	1.51	3.04	6.16
2000	1698	1.44	3.64	7.45	15.64	2000	1454	1.23	3.10	6.31	13.05	2000	1274	1.08	2.70	5.46	11.19
2500	2122	2.26	5.75	11.93	26.14	2500	1817	1.93	4.89	10.07	21.25	2500	1591	1.69	4.27	8.69	18.12
3000	2547	3.28	8.45	17.86	....	3000	2181	2.79	7.14	14.84	....	3000	1910	2.03	6.06	14.55	27.05

8-in. pipe

		Length of pipe in feet			
		1000	2500	5000	10,000
2000	954	0.34	0.85	1.71	3.46
3000	1432	0.77	1.92	3.89	7.97
4000	1909	1.37	3.46	7.06	14.77
5000	2386	2.15	5.46	11.29	24.51
6000	2864	3.10	7.98	16.83	....

		Length of pipe in feet			
		1000	2500	5000	10,000
2000	818	0.29	0.73	1.47	2.95
3000	1227	0.66	1.64	3.32	6.75
4000	1636	1.17	2.95	5.98	12.35
5000	2045	1.83	4.64	9.51	20.06
6000	2454	2.65	6.75	14.09	....

		Length in pipe of feet			
		1000	2500	5000	10,000
2000	716	0.26	0.64	1.28	2.57
3000	1074	0.57	1.44	2.90	5.87
4000	1432	1.02	2.57	5.20	10.65
5000	1790	1.60	4.04	8.23	17.61
6000	2148	2.31	5.86	12.05	26.16

10-in. pipe

		Length of pipe in feet			
		2500	5000	7500	10,000
1000	313	0.06	0.14	0.21	0.28
2000	627	0.27	0.56	0.84	1.12
4000	1254	1.11	2.25	3.37	4.56
6000	1881	2.52	5.15	7.73	10.62
8000	2508	4.54	9.37	14.06	20.06

		Length of pipe in feet			
		2500	5000	7500	10,000
1000	268	0.05	0.12	0.18	0.24
2000	537	0.24	0.48	0.72	0.95
4000	1075	0.95	1.92	2.88	3.88
6000	1612	2.16	4.38	6.57	8.95
8000	2150	3.86	7.82	11.73	16.54

		Length of pipe in feet			
		2500	5000	7500	10,000
1000	235	0.05	0.11	0.16	0.21
2000	470	0.21	0.42	0.63	0.84
4000	941	0.84	1.68	2.53	3.38
6000	1411	1.89	3.81	5.76	7.75
8000	1881	3.38	6.87	10.46	14.17

12-in. pipe

		Length of pipe in feet			
		2500	5000	7500	10,000
2,500	530	0.17	0.35	0.52	0.68
5,000	1061	0.70	1.41	2.11	2.82
7,500	1591	1.59	3.15	4.73	6.45
10,000	2122	2.83	5.77	8.66	11.95
12,500	2652	4.45	9.19	13.79	19.59

		Length of pipe in feet			
		2500	5000	7500	10,000
2,500	455	0.15	0.30	0.45	0.59
5,000	910	0.62	1.21	1.81	2.42
7,500	1365	1.35	2.69	4.03	5.53
10,000	1820	2.40	4.89	7.34	10.05
12,500	2275	3.79	7.76	11.64	16.19

		Length of pipe in feet			
		2500	5000	7500	10,000
2,500	398	0.13	0.27	0.39	0.53
5,000	796	0.53	1.05	1.58	2.11
7,500	1193	1.18	2.37	3.59	4.81
10,000	1591	2.11	4.27	6.46	8.69
12,500	1989	3.32	6.73	10.25	13.88

Computations in this line do nor invite, nor hardly permit, micrometrical precision, and refinements are out of place; hence it is quite permissible and very convenient to use 15 lb. for the normal atmospheric pressure, and this has been done in computing the tables herein given. A single example of the process will suffice.

Let there be 5000 ft. of 8-in. pipe, through which it is desired to transmit 4000 cu. ft. of free air per minute at an initial pressure of 105 lb. gage, or 120 lb. absolute. What will be the terminal pressure and the loss of head? The pressure here assumed is not unusual in the best practice of the present day.

Note that d^5 (8^5) is 32,768, and p^2_1 (120^2) is 14,400. Substituting these values the statement and solution is as follows:

$$\frac{4000^2 \times 5000}{2000 \times 32768} = 1220 = p^2_1 - p^2_2 = 14{,}400 - p^2_2$$

Then,

$$p^2_2 = 14{,}400 - 1220 = 13{,}180$$

and

$$p_2 = \sqrt{13180} = 114.80 \text{ lb.}$$

absolute terminal pressure; hence the loss of pressure is,

$$120 - 114.80 = 5.20 \text{ lb.}$$

The foregoing example shows what may be considered as near the highest permissible rate of pipe transmission, or a flow which should not be much exceeded in practice. The free air in this case being 4000 cu. ft. per minute, and the initial absolute pressure being 8 atmospheres, the actual volume, assuming that aftercoolers are used and that the air is at normal temperature, will be only 500 cu. ft. per minute. The volume content of an 8-in. pipe is 0.349 cu. ft. per foot of length; therefore the rate of flow will be

$$500 \div 0.349 = 1432 \text{ ft.}$$

per minute. A handy *limit* figure to keep in mind is the round number 1500 ft. per minute.

The loss of pressure will be a little more than proportional to the squares of the volumes of free air. That is, if in this case the volume of free air had been doubled, making 8000 cu. ft. instead of 4000 cu. ft., the loss of head would have been 22.44 lb. instead of $5.20 \times 4 = 20.80$ lb.

It is worth while to note how the pressure loss is diminished as the pressure is increased, due to the reduction of volume. Thus, in 1000 ft. of 1-in. pipe, transmitting 50 cu. ft. of free air per minute, the diminishing pressure losses for increasing initial pressures would be as follows:

Initial pressures	Loss of pressure
45-lb. gage	11.52 lb.
60-lb. gage	8.86 lb.
75-lb. gage	7.24 lb.
90-lb. gage	6.14 lb.
105-lb. gage	5.33 lb.
120-lb. gage	4.71 lb.
135-lb. gage	4.23 lb.
150-lb. gage	3.83 lb.

Of course, no tables can be compiled which will cover all the various requirements of compressed-air practice, but the figures herein given may at least furnish a working idea of the probabilities, and may be of service in a general way in preliminary estimates, or may serve to detect errors or inconsistencies which are apt to occur in the most careful figuring. No precise agreement with actual practice can be expected, as conditions which affect the result are so numerous.

The condition of the pipe itself no formulas can make allowance for. The actual diameters of wrought-iron or steel pipe, especially in the smaller sizes, are different from the nominal diameters. Some pipe is smooth and some has seams and blisters. The pipe may be straight or it may have numerous crooks and some elbows. As the computations of pressure losses usually have to do with transmissions to considerable distances few elbows are likely to occur, so that they seldom have to be taken account of, and if we say that an elbow gives a resistance equal to an additional length of pipe, we will probably come as near to it as any arbitrary formula could inform us.

No table or formula can take into account foreign substances or obstructions in the pipe, and it should be unnecessary to advise blowing out the pipe before it is put to use. If pipes follow the irregularities of the ground there may be low places where water will accumulate and interfere with the free flow. Water if it cannot be kept out of the pipe should be got out, and these low places offer the opportunity for draining, as spoken of elsewhere.

Strength of Pipe.—There is generally little question as to the strength of standard piping for the air-pressures generally employed, say up to 8 atmospheres or 105 lb. gage, but for the higher pressures special piping may be required. A generally accepted formula for computing the bursting point of pipe of given thickness is:

$$P=\frac{2I\times S}{D}$$

in which P = bursting pressure, gage
I = thickness, inches
S = tensile strength of material.

What is the bursting point of 10-in. wrought pipe 0.366 in. thick, internal diameter 10.019 in., tensile strength of metal 50,000 lb.?

$$\frac{2\times 0.366\times 50000}{10.019}=3653.$$

The above formula may be transformed to find the thickness of metal:

$$I=\frac{P\times D}{2S}.$$

What thickness of metal is required in a pipe 12 in. in diameter to burst at 3500 lb., tensile strength of metal 45,000 lb.?

$$I=\frac{3500\times 12}{2\times 45000}=0.466\text{ in.}$$

The tensile strengths in the above formulas may be considered safe in practice, the larger one being for steel.

Liberal factors of safety should be used, say 5 for air when there are no shocks or excessive temperatures to be encountered, but the factor when adopted should be held imperative.

Crane & Company, Chicago, did a good public service in testing some samples of pipe to the bursting point.

10 in. standard wrought iron burst at 1900 lb., while the rule gave 2922 lb.

10 in. extra strong wrought iron burst at 2700; by rule, 4102 lb.

None of the pipes burst at the weld, the rupture in each case being some distance from it.

CHAPTER XVII

RE-HEATING COMPRESSED AIR

It is sufficiently well known that after the transmission of compressed air to the point where it is to be employed, it having lost its heat of compression on the way, or, indeed, having had that heat abstracted by aftercooling devices, a considerable saving in the cost of the available power may be effected, theoretically at least, by re-heating the air before it is put to do its work. We still, however, have little reliable practical data concerning this suggested economy, or actual knowledge of the precise conditions under which it becomes available.

The specific heat of air being low, comparatively little heat is required from an external source to raise the temperature of compressed air rapidly and, under constant pressure, to increase its volume correspondingly. While air at a low temperature has a comparatively small cooling effect upon water, or upon whatever may come in contact with it, the fact inversely applied may be taken advantage of in the use of air as a power transmitter.

It may easily be shown that where a certain volume of air has been compressed to any given pressure, and has, by aftercooling, by transmission or by storage, returned to approximately its normal temperature, if that air is then re-heated and thereby expanded, the additional volume of compressed air resulting is produced by a much lower expenditure of heat than the original volume of compressed air was produced for, and also by a much lower expenditure of heat than would be required to produce an equal working volume of steam. The actual figures in the case, all theoretical, are as follows:

Weight of 1 cu. ft. of steam at 75 lb. gage = 0.2089 lb.

Total units of heat in 1 lb. of steam at 75 lb. from water at 60° = 1151.

Total units of heat in 1 cu. ft. of steam at 75 lb. = 1151 × 0.2089 = 240.44.

To produce by compression through a steam-actuated air-compressor 1 cu. ft. of compressed air at 75. lb and 60°, about 2 cu. ft. of steam of the same pressure are required, or the heat-units employed in producing 1 cu. ft. of compressed air will be about $240.44 \times 2 = 480.88$ heat units as the thermal cost of 1 cu. ft. of compressed air at the above temperature and pressure. The temperature and volume of the air as it leaves the compressor will be considerably higher than the figures here assumed, but as the air is invariably stored for a time, or is transmitted through pipes to a distance between its compression and its ultimate employment, it may be said to always return to its normal temperature before it is used, so that, whatever we may have at the compressor, the air as it is delivered to the motor, or whatever apparatus may be operated by it, will have cost, as above stated, 480.88 heat units for 1 cu. ft. at 75 lb. The difference in the thermal cost of any volume of compressed air thus produced by mechanical compression and the cost of any additional volume of air that may result from the subsequent re-heating of the air is very striking.

The weight of 1 cu. ft. of free air at $60° = 0.076$ lb.

Weight of 1 cu. ft. of compressed air at 75 lb. and $60° = 0.456$.

Units of heat required to double the volume of 1 lb. of air at $60° = 123.84$.

Units of heat required to double the volume of 1 cu. ft. of compressed air at 75 lb. and $60° = 123.84 \times 0.456 = 56.47$.

Cost of 1 cu. ft. of superheated compressed air at 75 lb. compared with the cost of 1 cu. ft. of compressed air as produced by ordinary compression:

$$480.88 : 56.47 :: 1 : 0.1174.$$

Here we see that the cost in heat-units of the volume of air produced by the re-heating is less than one-eighth of the cost of the same volume produced by compression. Upon this showing the matter is certainly worth looking into.

The operation of re-heating compressed air is correctly so termed. It is, in fact, a case of doing work over again, or of replacing in the air heat that has been lost by it in previous operations. It must be confessed that the presumption is all against our finding much profit in this direction. There are not many places in life where it pays to do our work a second time. There is, as we have seen, practically no air compression without heating the air by the operation, and there is no transmission of air after compression without its cooling to very near its original temperature. If the air could go immediately from the compressing cylinder into the motor cylinder, where it does its work, without losing any of its heat, it would have the same effective power as it would have after long-distance transmission and cooling and re-heating, and without the additional cost of that re-heating.

While we are saying in all good faith that there is little loss of power in the transmission of compressed air to considerable distances, and that the difference in the pressure of the air at the two ends of a long pipe necessary to overcome the friction and maintain the flow is but small, and that it is to a great extent compensated for by the increased volume at delivery, the fact still is that there is a great loss of power in the transmission of the air, if we reckon from the moment when compression ceases, on account of the inevitable cooling of the air. Still this loss is not properly chargeable to the transmission, for no matter how far the air may be transmitted the cooling is all accomplished before the air has travelled very far, if the pipes are of proper size. Supposing air to be transmitted 10 miles, it must be conveyed with considerable rapidity if it does not get down to normal temperature before the end of the first quarter of a mile.

As the volume of air under any constant pressure varies directly as the absolute temperature, it follows that to double the volume by heating the air its absolute temperature must be doubled. The air being at 60°, its absolute temperature will be $60+461=521$, and double this will be $521\times 2=1042$, the absolute temperature required. This by the thermometer will be $1042-461=581°$. As this is the temperature that is required for the air when delivered into the motor, and actually beginning its work, it will be necessary, on account of the ease and rapidity with which it cools, to heat the air considerably higher than this theoretical temperature. It is one thing, and an easy one, to heat the air, while it is a very different and a very difficult thing to keep it hot. To avoid all loss of heat it would be necessary, not only to keep the pipe which conveyed the air constantly hot, but also the cylinder in which it was used, or it would be cooled before it began to do its work.

In one case within my experience, where compressed air was re-heated, and its absolute temperature was increased at the heater 38 per cent., and where, of course, its theoretical increase of volume was the same, the actual increase of power realized was only 12 per cent. In this case the air was transmitted after the re-heating about 20 ft., the pipe was not covered, and no precautions were taken to prevent loss of heat by radiation. The volume of air transmitted was sufficient to develop between 20 and 30 h.p. The theoretical augmentation of temperature required to double the volume of compressed air at 60° being 581°, the actual temperature required at the heater under the most favorable conditions in order to have a double volume of air available *in* the motor will not be less than 800°, and this is a temperature that it is practically impossible to employ and maintain, lubrication troubles would defeat it if nothing else, and we may as well give up all thought of doubling the volume of compressed air by re-heating it and of realizing the promised economy of such an operation.

If instead of doubling the volume we only attempt to increase it by one-half, or 50 per cent., which it is practicable to do, the required theoretical temperature (absolute) will be 521+50 per cent. = 782, and 782 − 461 = 321°, the sensible temperature required. Adding enough to this to allow for the intermediate cooling, the actual temperature required should probably be not less than 450°. The temperature of the air before the re-heating being assumed to be 60°, the increase of temperature will be 450° − 60° = 390°. As we saw above that it required 56.47 heat units to raise the temperature of 1 cu. ft. of compressed air at 75 lb. gage pressure from 60° to 581°, the actual increase of temperature being 581 − 60 = 521, it follows that to raise the temperature 390° will require:

$$521 : 390 :: 56.47 : 42.27.$$

Then if the first cubic foot of compressed air costs 480.88 heat units for its compression, and if the additional half of a cubic foot produced by re-heating costs 42.27 heat units, the total cost of 1 1/2 cu. ft. under the re-heating system will be 480.88 + 42.27 = 523.15, and the cost per cubic foot at this rate will be 523.1 ÷ 1 1/2 = 348.76 heat units. The relative cost in heat units of 1 cu. ft. of compressed air produced by compression alone, and of a cubic foot resulting from compression and re-heating, will be:

$$480.88 : 348.76 :: 1 : 0.72.$$

From this it appears that the gain by re-heating compressed air sufficiently to increase its effective volume 50 per cent. will be 28 per cent. The more fair and correct way to state this, however, will be to reverse it:

$$0.72 : 1 : : 1 : 1.38.$$

We may say, then, that the total fuel applied with the re-heating system will yield 38 per cent. higher results than are to be realized without the reheating. This seems to be very near the maximum that can be attained in the way of economy by re-heating *dry* compressed air.

But, after all, it must be confessed that it is not always, nor indeed often, that the re-heating of compressed air is practicable or possible. Bearing in mind the facility and rapidity with which heated air in transmission loses its heat, it is idle to think of ever

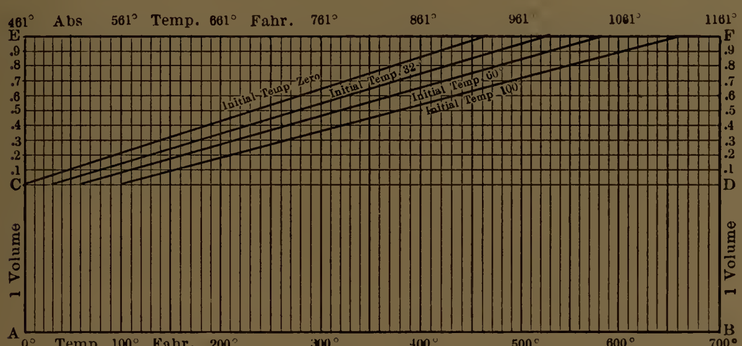

FIG. 52.—Increase of Volume by Reheating.

profitably re-heating compressed air except for continuously running motors, and then by heaters very close to the motors.

Wherever any motor or engine is to be run by compressed air without interruption, a heater for the air should certainly be employed; but there are now several hundred different and distinct uses of compressed air in not one of which it would be practicable or anything but a losing operation to try to heat the air.

Fig. 52 shows the increase of volume accompanying the heating or reheating of compressed air. The air is assumed to be heated from the several initial temperatures of 0°, 32°, 60°, and 100°, the pressure remaining constant during the operation represented. The relative volume at any temperature is indicated by the height of the vertical line corresponding with that temperature, the height from *AB* to *CD* representing one volume, and each horizontal line above that indicating, successively, an additional one-tenth of volume. When the line *EF* is reached, the original volume is doubled. Figures below the base-line *AB* indicate the sensible temperatures Fahrenheit, and the figures above the upper line indicate the corresponding absolute temperatures.

Fig. 53 shows the increase of pressure only caused by the heating of compressed air, the volume being constant. The air is assumed to be heated, as in Fig. 52, from the several initial temperatures of 0°, 32°, 60°, and 100°, and also from a number

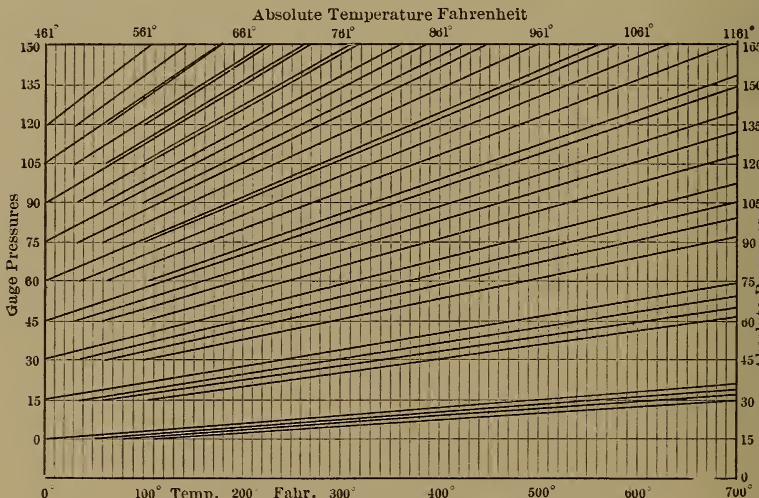

FIG. 53.—Increase of Pressure by Reheating.

of different initial pressures. The pressures are indicated by the several horizontal lines, the vertical distance between any two adjacent lines representing approximately 1 atmosphere. The figures at the left of the diagram indicate the gage pressures, and the figures at the right the absolute pressures. The temperatures are indicated as in the previous diagram.

It looks as if the most unsatisfactory phase of the re-heating problem to-day is in the re-heater itself. There are re-heaters so-called of various makes on the market, but where is *the* re-heater? It is not enough to provide a receptacle where fuel may be burned and then sufficient heating surface over which, or heated tubes through which, the air to be heated may be passed. This might do when the air was flowing steadily all day long, as for instance in driving a mine pump, but where else of all the places in which compressed air is used?

Supposing the heater to be of the right capacity and the fire adjusted to give the required air temperature when flowing at a certain rate, then if the air flowed faster it would not be heated enough, and if it flowed slower it would be overheated. If the flow stopped entirely and then after a time started again the air might be so hot as to burn the packing or make other trouble in that way. In either case there would be waste of fuel and the economy sought would be only partially realized, if at all.

Then with this arrangement who is to know what the actual temperature of the air is, except by watching a recording thermometer and then only to realize that the heat is not right and is not steady at any figure? The proper working temperature for the air should be decided upon before hand, and then the ideal reheater should maintain this temperature constant, within narrow limits, just as an intercooler or an aftercooler maintains an approximately constant temperature when working in the opposite direction. These coolers deliver the air at a temperature approximating that of the circulating water whose temperature varies only imperceptibly; so it would appear that if water were employed as the circulating medium, and if this water were kept at a constant temperature then the temperature of the air heated by it could be depended upon to be nearly constant.

The temperature of the water in a steam boiler rises or falls with the steam-pressure, and if the steam-pressure remains constant the water temperature coincides with that of the steam —or rather *vice versa*. By carrying a proper steam-pressure the right water temperature for re-heating compressed air may be obtained and maintained; and this steam pressure need not be maintained with minute precision, as the temperature changes are not rapid. Thus with 100-lb. gage pressure the steam and water temperature would be a trifle under 340° while at 200 lb. the temperature would be 50° higher, and anywhere within this range the air would be heated to above 300°.

A more constant steam-pressure and a steadier temperature may be maintained than here suggested, as is often done in house-heating arrangements, by means of automatic dampers and other fire-controlling devices, without blowing off or wasting the steam, or the hot water behind it, and all of the heat of the fire actually used would be that which went to the heating of the air.

The better arrangement would be to have a large tubulated steam space and a small water space and to heat the air by the steam rather than by the water. The smaller body of water would enable it to be heated and steam generated quicker, with better command of the temperature fluctuations.

It may be possible that there are other combinations equally effective for re-heating compressed air, but that here suggested has the advantage of having been successfully employed in at least one notable instance. At the admirable compressed-air installation at the mines of the Anaconda Copper Mining Company at Butte, Montana, where compressed air has supplanted steam for the driving of the big hoisting engines as spoken of in the preceding chapter, the air is re-heated by steam, and the additional power gained by the re-heating is stated to be obtained at a cost of one-third of a pound of coal per horse-power hour.

CHAPTER XVIII

COMPRESSOR AND RECEIVER FIRES AND EXPLOSIONS

It is a rather curious fact that the air-receiver, so far from being the air cooler which it is intended and assumed to be, has often been transformed into a combustion chamber and an air heater.

A trouble all too frequent in compressed-air practice comes from the lubricants used in the air-cylinder, a combustible residuum from which accumulates in the air passages, upon the valves and in the air-receiver, often taking fire and causing sometimes disastrous explosions.

Preceding or accompanying the deposition of this combustible residuum, the more readily volatilized constituents of the lubricants are liberated and mingle with or are carried along by the air. It is quite fortunate that there is a somewhat narrow limit to the relative proportions of air and of volatilized oil ingredients which will render the mixture explosive, a trifle too much or too little of the latter making the combination comparatively safe so far as instantaneous ignition and the sudden development of destructive force is concerned. When, however, the explosive proportions do occur, and the means of ignition also concur, then sudden and serious ruptures ensue with disastrous and often fatal results.

While apparently the explosive mixture, even if combined in proportions within the explosive limits spoken of, is not likely to take fire of itself, but awaits the spark or flame to fire it, the solid or near-solid carbonaceous deposits do take fire spontaneously when sufficiently high pressures and temperatures are reached, the flames generally spreading rapidly in the receiver and in the pipes beyond, such flames sometimes traversing entire lines of piping, even reaching to the drills or other air actuated apparatus in mine or tunnel.

This phenomenon was experienced several times in the work on the "New" Croton aqueduct about a quarter of a century ago. Quite recently a case was before the courts in the state of Alabama where a man working about a pump driven by air was asphyxiated by the products of combustion delivered by the air pipe and lost his life.

It has sometimes happened that such a flame as this has found somewhere in the pipes an explosive mixture of the right proportions and has fired it, with all the disastrous consequences of a real explosion.

This matter may best be understood from the example of an actual occurrence in this line. A small air-receiver connected with a portable gasoline-engine-driven air compressor employed on street gas-main work exploded on June 24, 1912, near the entrance to Greenwood cemetery, Brooklyn, the occurrence exhibiting a number of interesting features illustrative of the conditions under which such "accidents" may occur.

To save time it may be premised that circumstances, after the event, indicated that in this case there was no sudden and enormous increase of pressure, such as would have been caused by the ignition of an explosive gaseous mixture, such for instance as occurs in the gasoline engine and furnishes its driving force. The normal working pressure usually carried in the receiver was probably not much if at all exceeded, but the receiver, having but a minute factor of safety at the best, was temporarily weakened under the abnormal conditions which arose, and the head let go.

The receiver was suspended horizontally under the frame which carries the gasoline engine and the compressor. It was 6 ft. long and 2 ft. in diameter with dished heads, the convex side of the head being outward at one end and the concave side outward at the other end, this being the too familiar practice just for convenience in riveting in the last head, and for no other good reason. The air entered the receiver from the compressor about 2 ft. from one end of it and at the central height, and the air was taken out at the other side just opposite the inlet, see Fig. 54. A pop safety valve on the pipe leading from the compressor was set at 110 lb., and the usual working pressure was about 100 lb.

As disclosed after the explosion, the entire interior surface of the receiver had been covered apparently with all the oil it could carry. If the compressor runner had been told to use all the oil he could, instead of as little as possible, the surfaces probably could not have been more oily. It would have been proper of course to have drawn off from time to time all the oil and water which might have collected at the bottom of the receiver, and this may or may not have been done, but it evidently would not have made the surfaces clean or have freed them from the gummed or thickened oil residuum with which they were coated.

The condition of the inner surface (*A*, Fig. 54) of the head which blew out, the head with the convex face inward, was as different from this as could be imagined. It was dry and absolutely clean, and the color and condition of it showed that it had been red hot. It was evident that a local fire had raged, perhaps only for a minute or so, on this then oil-coated surface, burning the oil off and suddenly heating the plate, the fire not having had time to spread to the adjacent surface of the cylindrical shell, and then the explosion came.

In the act of explosion several curious things happened which

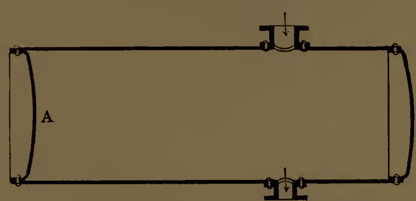

Fig. 54.—Section of Exploded Air Receiver.

it is worth while to note carefully. The head became so hot that, as it was under about all the air pressure it could stand anyway, the dishing of the head was suddenly reversed, and the originally convex surface became concave, as Fig. 55 shows, this being a view of the inner surface. That this is the inner surface of the head is corroborated by the lip at the edge of the sheet. The ridge which is dented across the face was evidently caused by something against which the head struck, presumably the axle of the machine, as it flew off.

We can readily imagine that this reversal of the dishing of the head took place with considerable of a snap, the shock and momentum, in addition to the aggregated air pressure, 45,000 lb., being sufficient to cause the head to let go.

When it did let go not a single rivet was sheared. It will be noticed in Fig. 55 that when the reversal of the dishing of the head occurred and the tearing-away-act was going on the

Fig. 55.—Head that Blew Out with Bulge Reversed.

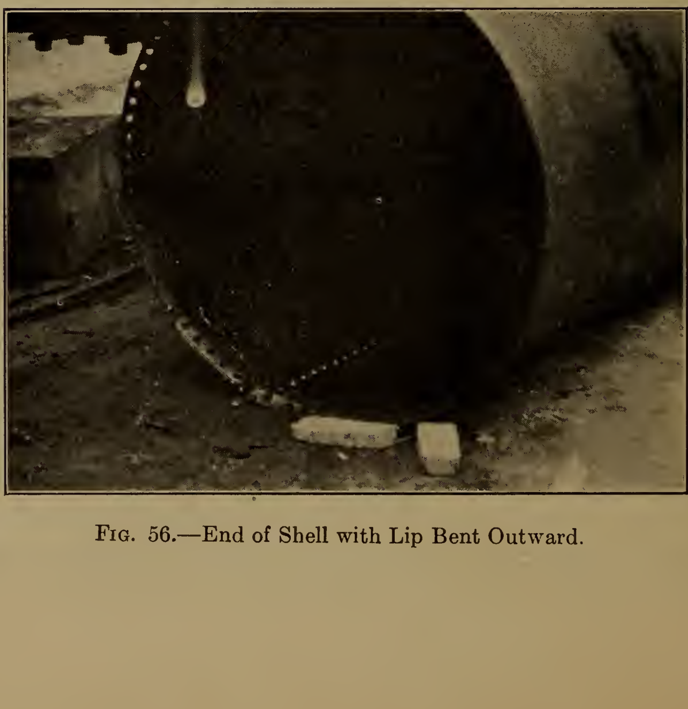

Fig. 56.—End of Shell with Lip Bent Outward.

rivet-carrying lip of the head was bent outward all the way around, so that the opposite sides, instead of being parallel, stood off 30 to 45 degrees from the normal position, the end of the cylindrical shell to which they were attached bending outward in the same way. See Fig. 56. There were 80 rivets, 3/8 in. diameter, and when the lip was bent out in this way these rivets just pulled their heads off, dropping them outside the shell all around, and then the rivets pulled out of the holes and the job was done. The fracture of the rivets was square across under the riveted heads and flush with the outside of the shell. Of the entire 80 rivets just 3 kept their heads on, and these took their leave of the shell by tearing notches out of it.

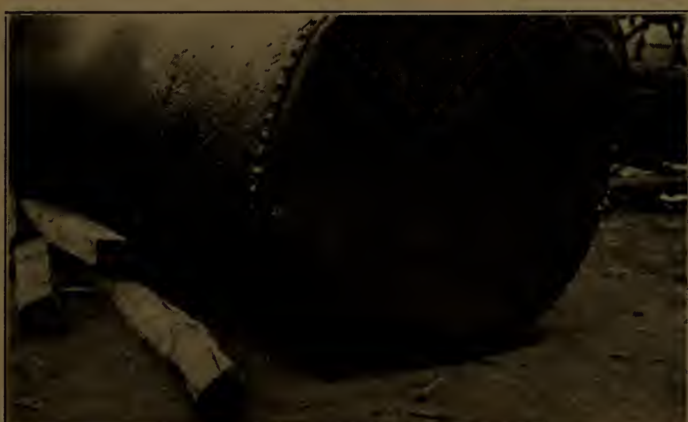

Fig. 57.—Other End of Receiver.

When the head flew off the body of the receiver naturally was driven in the opposite direction, but it could go only a few inches when it struck the axle of the machine, which dented the head as seen in Fig. 57. Besides this the shell seems to have received no other damage.

The explosion seems to have been such as might have been expected from the normal air-pressure, and was not comparable in destructive effect with what it would have been if the receiver had been filled with an explosive mixture, such as we utilize, for instance, in the cylinder of the automobile.

Undoubtedly receivers have often taken fire internally as indicated in this case, but having a sufficient factor of safety notwithstanding the heating of the metal, they have been able to stand the strain and no accident has resulted; nevertheless the occurrence is one which it is not pleasant to think of, and every precaution should be taken to prevent it. It is not too much to say that every receiver should be made strong enough to stand, without bursting, the sudden heating of the head if the oily deposit within should take fire, because that occurs so frequently. If a receiver is said to be tested cold by hydrostatic pressure to 165-lb. gage and then is guaranteed for a working pressure of 110 lb., we have only a margin of safety of 0.5, this with everything cold and quiet. With a part of the receiver nearly or quite red hot and subject to the working shocks of the machine the only proper thing to expect should be the annihilation of the factor of safety and of the receiver with it.

Only a few weeks after the occurrence here spoken of there appeared in *Power* an account of a fire of this character, but without explosion, in a vertical air-receiver which had the same kind of inwardly dished head at its lower end. After the fire got under way and the bottom of the receiver became visibly red hot the dishing of this head was reversed, so that the middle of the head bore on the ground, taking the entire weight of the receiver and lifting the projecting edges of it a couple of inches off the foundation.

Chinese Air-receivers.—It might not be inappropriate to call the receivers of the type here spoken of "Chinese" receivers. It is well known that the Chinese have never used barrels for the packing of their merchandise, and the reason that has been assigned for it is that they have never found a way of putting in the last head of the barrel without having a Chinaman inside to hold it up.

Are we not in the same fix with our air-receivers, especially those of the smaller sizes which are the most numerous? These receivers are practically all of the same pattern. They are made with the dished heads, which is eminently proper, and one head is placed with the convex face outward, which also is proper both for strength and looks, while the other head is set with the bulge of the head facing inward, a thing disgusting to any one with normal mechanical instincts, until he has become too familiar with it, and the only assignable reason for it is that they can't rivet the head in the other way without somebody inside to insert and hold the rivets.

Rivets Should be Banished.—But why use rivets at all? May not the day soon come when we will realize the inefficiency of, and the absurd waste of material in, the riveted joint, and may not later the time come when riveting will be abandoned and samples of it become an exhibit in the museums? The writer lately stood with a young lady friend beside a high-pressure steam-boiler where the rivets were a protuberant feature. "How strong that boiler must be with so many big rivets in it," she remarked. "Why, my dear girl, every rivet represents a hole in the sheets and that part of the boiler is the weakest instead of the strongest part of it." This, of course, is always true. With double the thickness of metal and the added weight of the rivet heads the joint is the weakest part and weakens the entire structure by 25 or 30 per cent. And that is not all. The sheet is 25 or 30 per cent. thicker over its entire surface, and the same percentage heavier, than it need be to equal the strength of the joint.

In our chapter on air transmission the welding of pipe lines is spoken of as having been successfully practised. It would seem that the practice should be equally applicable to air-receivers. The receivers are perhaps the most unsatisfactory detail of compressed-air practice to-day, and getting rid of the rivets and making receivers of uniform strength throughout should be a desirable achievement. The welded receiver should be stronger for the same weight, or lighter for the same strength, but more important would be the possibility of placing both the heads correctly, always assuming that a welding process is employed which can be operated entirely from the outside, as the pipe line spoken of was welded.

How Receivers take Fire.—Every precaution should be taken to prevent the accumulation of oily deposit in the air-receiver, and not only should the receiver be drained at frequent and regular intervals, but it should also be examined and cleaned out at appointed times whenever the construction will permit. Supposing that there is an accumulation of water and oil in the receiver, the draining process may take out the water but leave the floating oil to finally cling to the surfaces which it may come in contact with as the water is leaving it.

The interesting question remains as to the conditions causing or accompanying these ignitions of the oily surfaces, and there is usually more or less of mystery made of this phenomenon when it occurs. My thoughts carry me back to what was one of the familiar operations of the shop when I was every day in it, and that was the oil-tempering of steel springs. The spring is first heated in the fire to a bright red and then is plunged into oil and cooled. This leaves it hard and brittle and it is drawn to the proper temper by "blazing off." To do this the spring while dripping with oil is held over the fire, which must be without flame, and is slowly and carefully heated until the oil on the spring bursts into flame. The oil is not ignited by the fire. There is no flame or spark from the fire that does it; the flame simply comes of itself, as we might say, when the spring reaches the right temperature. When the spring reaches this temperature the oil will keep burning until the oil is burned off, or the spring while blazing may be dipped in the oil and the flame will go out, but if the spring is quickly drawn out of the oil so that it is not cooled much, the flame will "light itself" again as before, without being held to the fire at all. There is a certain temperature at which the oil will thus ignite spontaneously, and this temperature is so fixed, differing, however, for different oils, that it has been taken for generations as an index of the heat required for the tempering of springs, swords and other responsible articles of steel.

Precisely the same thing that happens in the oil tempering of springs will happen to the oil-coated surfaces of air-receivers, pipes, etc., when they get hot enough. The oil will take fire of itself without the impulse of any spark or flame or other extraneous means of ignition. The oil-covered steel spring when the oil upon it takes fire in the open air is below a visible red heat, and that begins at about 700°. In the air-receiver whose explosion we have been considering the temperature due to compression may have been as high as 500°. With air when compression begins at a temperature of 60°, the theoretical terminal temperature after adiabatic compression to 105 lb. is 496°. This compressor was running in bright sunshine on a hot day and the intake air passed close to the already heated air-receiver before entering the cylinder, so that its temperature at the beginning of compression was presumably above 100° and its terminal temperature above 500°. The fact that at this end of the receiver there was nothing to cause any active movement of the air in contact with the oily surface may have offered special opportunities for the atoms of carbon and of oxygen to make love to each other and consummate their union.

We must remember that with the air at 105 lb. gage, it is compressed to 8 atmospheres, which means that the molecules —if that's the word—of carbon in the oil are individually exposed to or in contact with 8 times the quantity of oxygen that they would be in contact with if the air had only the normal pressure of 1 atmosphere, so that we may readily believe not only that the oil would be ready to spontaneously ignite at a lower temperature but that it would also burn more fiercely after the ignition.

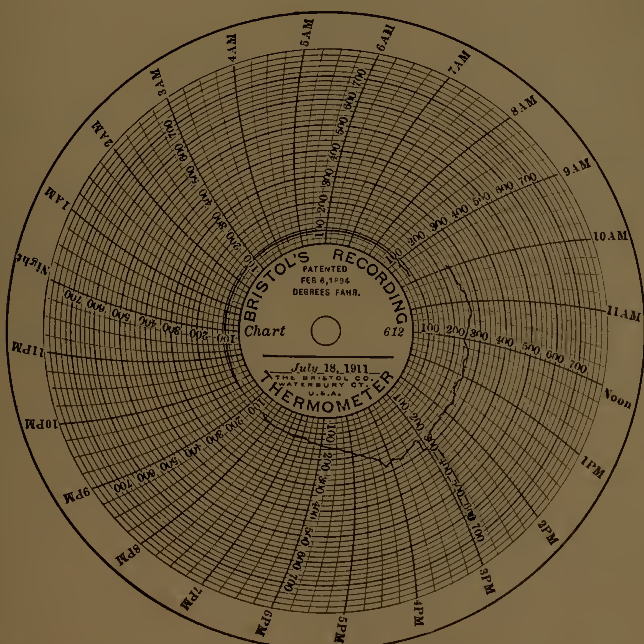

Fig. 58.—Thermometer Record of Air Receiver Ignition.

There recently appeared in *Mines and Minerals,* reprinted in *Compressed Air Magazine,* June, 1912, an account of an ignition of this character, but without damaging explosion, in connection with a four-stage compressor employed for charging mine locomotives. The ignition occurred after the fourth stage of the compression, when the air was at a pressure of 65 atmospheres, and a recording thermometer which was providenially attached and in operation showed that the ignition took place at 275°, which seems to corroborate our suggestion that the higher the pressure the lower will be the temperature at which spontaneous ignition will occur.

The thermometer record spoken of is reproduced in Fig. 58. The record with the compressor in operation begins at 10 A. M. The normal temperature when working before this had been 240°. At 11 A. M. the temperature began to rise a little and at 3 P. M. it was close to 280°. Then the oil stuff took fire and in a minute or two the temperature was above 600°. There was a fusible plug in the receiver or its connections which melted and blew out. The compressor was stopped, the fire went out at once for lack of fresh air and the temperature dropped back to where it was before. As there was immediate call for the air the compressor was started again at once and ran until 4 P. M. The remainder of the record is of no account as it was made after the stoppage.

As to Quantity of Lubricant—No formal rules can be established as to the quantity of air cylinder lubricating oil that should be used in any given case, as the conditions may be so different, both as to the oil itself, the design and arrangement of the machine, the speed at which it is run, the intake temperature of the air, the efficiency of the jacket cooling, etc., but one or two suggestive examples may be cited.

A report was published in *Power* in 1911 of a month's running of the air compressors on the Panama Canal, and in that report it was shown that at Rio Grande a gallon of air cylinder lubricating oil was used for 3,360,124 cu. ft. of free air compressed; at Empire it was a gallon of oil for 4,317,716 cu. ft.; and at Las Cascadas for 5,163,936 cu. ft.

It has only recently been reported that a compressor which has been running in the power house of the Nevada Consolidated Copper Company, 24 hours per day for five years past, is compressing 16,000,000 cu. ft. of free air per gallon of air cylinder lubricating oil, this rate of oil consumption being less than one-third of that at Las Cascadas, cited above.

To get an idea of how this rate of oil consumption works out we may say that a compressor with an air cylinder 24 in. in diameter, which is quite a common size, and a piston speed of 400 ft. per min., will compress 1200 cu. ft. of free air per min., or 1,728,000 cu. ft. per 24 hours. Then the oil used per day by this compressor, at the rate of 1 gallon per 16,000,000 cu. ft., would be only 0.108 gallon, or less than a pint for the 24 hours.

In this run of 24 hours the cylinder surface swept by the piston would be 83 acres, and continuing the run until a pint of oil was consumed the cylinder area traversed would be 96 acres. It is safe to say that at this rate the accumulation of oil in the air receiver or pipes would not be very rapid.

In sharp contrast with the above careful use of oil in air cylinders we have the following from a correspondent of the *South African Mining Journal:*

"It has happened to me scores of times that I have had to leave the machines and come to the surface and ask the man in the power house to give the air receiver a tap, because the air has been so bad from the compressor oil, making everyone ill. I have known one mine where this used to occur regularly every week, the miner and all his boys suffering from bad headaches, caused by the gas in the air. I complained to the management on three occasions, and finally an investigation was made, and there were taken out of the air receiver 110 *gallons of compressor oil.*"

CHAPTER XIX

SIDE LINES FOR THE AIR-COMPRESSOR

There are "side lines," or incidental employments for the air-compressor which are somewhat outside of its strictly legitimate line of work and which add considerably to the range and volume of its business. Air-compressors, without change of design or construction find frequent, extensive and constant employment in the compression of the various gases other than air. In fact compressed air and compressed gas distinctively so-called, whether it be natural gas compressed by the forces of nature, or manufactured illuminating gas artificially compressed, with other elementary gases which require manipulation or transfer in the chemical and other industries, are so similar in so many of their physical characteristics that it would seem to be more or less an ignoring of the functions and the adaptabilities of the air-compressor not to speak of its services, actual and potential in this field. For the air-compressor to be offering, and indeed urging the acceptance of its services for gas compression would seem to be natural enough, and it also would seem that it has some right to feel slighted that its proffered services are not more generally accepted, and that the greater gas interests give it so little to do when there is so much that it could do so well for the benefit of all.

The Compressor and High-pressure Gas in Cities.—The substance of what follows for a considerable portion of the chapter appeared some time ago as a personal contribution in a leading engineering journal of New York, and it is reproduced here with little change except a few additions. It needs not to be said that it deals with a matter which is not to be decided without thorough consideration and perhaps extensive discussion. It took shape in the mind of the writer without any recognized personal suggestion from any source, and it was submitted to the editor absolutely without the knowledge of any third person. So much for the responsibility; if in any particular the writer errs; or if he is wrong all through, there are those who can and who should tell him so.

By what Right?—The title of the article was originally a question: "By what Right?" and it was intended to fairly characterize what it introduced. The attitude of the writer upon this topic was, and is, interrogatory rather than assertive, although the latter may seem to be most suggested. It is expected that it will appear to be high time for some one to be authoritatively propounding the question here crudely suggested.

The simple question is as to the permissible retention of the ancient methods of gas storage and distribution, with special reference to the protuberant gas-holder, and this from the viewpoint of neither the gas producer nor the gas consumer, as such, but of the general and long-suffering public.

Gas, of course, is an established necessity to practically all the people, and all questions as to cost of production and distribution, quality of gas furnished and convenience and reliability of service are to be settled between producer and consumer, with or without the aid of legal enactments, and no one else so far is interested.

It happens, however, that the method of storing and distributing the gas cannot be indifferent to the otherwise disinterested public, for it touches all at more than one sensitive point, and in an objectionable way which should not be tolerated or permitted, except in so far as it may be unavoidable. We have been so familiar for so many years with the sight of the supreme uglifier in every large outlook in every city in the land that we do not realize the unsightliness of it; we do not think to protest against it, or, in fact, in any way to question its presence. Who has thought of asking by what right the gas-holder intrudes, or has suggested its expulsion if its necessity and right are not proven and upheld?

The question is so far from ever having been formulated that the gas-holder has never treated the public as in any was entitled to an explanation or a justification wherever and whenever it has chosen to plant itself. It has no doubt at times had to establish certain legal rights to locate, but always upon the unquestioningly conceded assumption of the imperative necessity of it. Is it so necessary and indispensable? If so, it should be "up to" the gas people to prove it in the light of the present century. When it came to proving the necessity of the telegraph poles they quickly fled the city streets.

Few realize how bad the case is, or, indeed, have given the matter any thought at all, and it would seem to be an opportune time to stir things up. Civic pride is becoming alert and restive. We are beginning to take an interest in the appearance and condition of our cities, and many movements are on foot for their betterment. But what shall we do with the gas-holder? Think of the costly viaduct starting from Grant's Tomb in New York and connecting to the upper stretch of the beautiful Riverside Drive all completed, and then almost immediately the popping up of the afreet we see in Fig. 59.

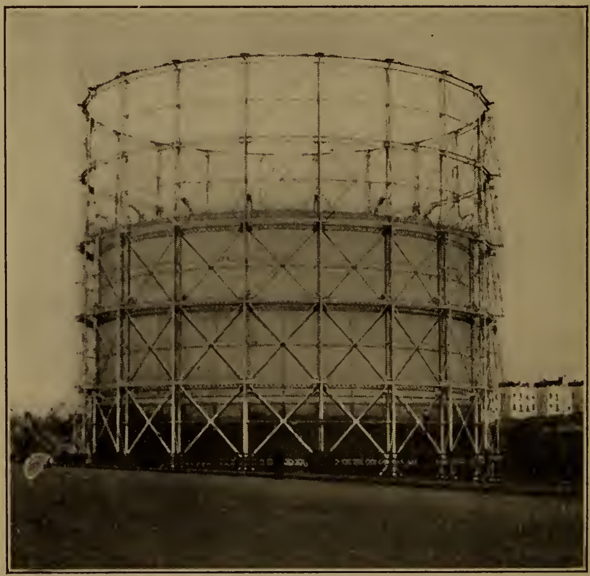

Fig. 59.—By What Right?

No one objected or thought of protesting at the time or since, as far as we have heard, because it is a gas-holder, you know.

The work of redesigning, rearranging and permanently beautifying our cities, of rendering them more satisfactorily habitable, so that not only we, the indwellers, but also the incomer and the transient onlooker, shall say it is good to be here, cannot proceed far before we realize that much of our doing must first of all be undoing. We cannot re-arrange and upbuild to our liking until we tear down, banish or obliterate the things which, if undisturbed, would render our efforts futile. Objectionable and, but for our familiarity with them, often disgusting features have accumulated and established themselves unchallenged, and yet if they are allowed to remain there can be no real progress toward permanent and satisfactory improvement.

FIG. 60.—Looking up West End Avenue.

In Fig. 60 we are looking up West End Avenue, New York, a beautiful and high-class residence street which retains its select character all the way up to the end of it, two or three miles to the

FIG. 61.—Looking up West End Avenue.

north. In Fig. 61 we are still looking up West End, Avenue but from a point just a quarter of a mile further down. From the same point, turning to the right, we have Fig. 62, a row of well-built tenements, but only colored people can be found to occupy them. Fig. 63 was their outlook when this was written. Since then these unimproved lots, there being apparently no prospect

FIG. 62.—Tenements facing the Gas Holders.

of erecting respectable, substantial, permanent buildings upon them, have mostly been covered with cheap and shabby sheds

FIG. 63.—The Outlook from the Tenements.

for storing carts, etc., which city ordinances do not permit to stand in the street at night. Directly opposite these lots, on the other side of West End Avenue, are the primitive rocks of

Manhattan, with squatter shanties surmounting them, neither of which, shanties or rocks, it has been worth while to remove. The next block to the north on the same side of the avenue is a lot, without buildings, in which castings and steel work are stored.

Farther away in all directions, for, say, three or four blocks all around these gas-holders, they have been the means of accomplishing, as some might say, a work of great beneficence by so depreciating the property values as to make possible the erection all through the neighborhood of tenements of the cheapest class for the occupation of the minimum wage-earners and of the strugglers for precarious subsistence. If it were not for the blessed gas-holders where would these poor people go?

This is not in the outskirts of the city, but in the heart of it, the location being in the Sixties, while the island is solidly built up for more than a hundred blocks above. There is everything to warrant the presumption that all this section of the city, of which this one group of gas-holders is the center, would be very differently occupied and improved if the gas-holders were not there. Certainly it would all be well and profitably used, which it is not now, and the location, otherwise desirable and easily accessible, deserves a better fate.

We could get pictures of similar character to those here presented from each of the dozen or so of gas-holder neighborhoods of Manhattan, and the same of every other large city, showing them all to be nuclei of desolation and responsible for the depreciation of property values amounting in the aggregate to hundreds of millions of dollars. It is not for the present writer to estimate the amount of this depreciation, but it would be well for real estate experts to be doing some figuring upon the problem.

Suppose that some day there should come to some one the assurance in advance that the gas-holders in the cities would all have to go (and the gas companies are likely to be themselves the first to realize it), what an attractive and promising speculation it would be to quietly buy up all the depreciated property in these gas-blighted neighborhoods.

The gas-holder is simply to-day the survival of the unfit, if not of the unfittest, and it seems more tenacious of life than any other thing of which we have record. Nothing can be more certain than that *if the gas business were beginning as a new business to-day it would not begin with the absurdly low-pressure service* *now in use and it would not use the big gas holders;* but it began in that way a hundred years ago and has not changed.

> "Little of all we value here
> Wakes on the morn of its hundredth year
> Without both feeling and looking queer,"

and low-pressure gas is queer enough.

Just think of it. Ordinary city gas is transmitted and stored and distributed at pressures so minute as not to be measurable in pounds to the square inch, as we commonly measure and record pressure, or even in ounces, but in tenths of an inch of water.

We happen to have conveniently at hand the figures from a typical city plant. At Syracuse, N. Y., they have about 170 miles of gas mains, from 2 in. to 20 in. in diameter, and the gas pressure varies from 1 1/2 in. to 3 1/2 in. of water, these being the limits, or, say, 1/20 to 1/8 lb. to the square inch. In the literature of the gas men the maximum pressure here mentioned is spoken of in the record as 37 tenths of an inch of water. Why, a boy with a tin bean blower could give you double that pressure. A familiar boy's trick is to blow into a burner against the city gas-pressure, filling the pipes with air and putting out the lights.

And yet, the existence of the gas-holder is absolutely conditional upon the retention of these low pressures, these pressures of a hundred years ago, in storage and in distribution. Any, even a slight, increase of pressure would be death to the gas-holder at once. Take any one of the largest gas-holders, such, for instance, as the first one shown here, and an increase of 1 lb. in the pressure within it would require an addition to the weight on the top of about 2000 tons. This would give a steel top over 3 in. thick, an effective armor against aeroplane bombs.

It would, in fact, be impossible, for another reason, to carry an additional pound of pressure in the gas-holder, even if it could be weighted down sufficiently. In all candor and seriousness, the modern gas-golder is a magnificent achievement in engineering, and one of the wonders of it is the telescoping feature. When the holder is full and has risen to the top of its guides it is not, as it looks, a single shell, but consists of four or five "lifts," which slide into each other as they descend. To make a gas-tight joint between the lifts there is to each a "water seal" which retains the gas with absolute security, as long as it holds it at all, but if the gas pressure were increased to 1/2 lb. to the square inch or about that, instead of 1/8 lb., the present maximum, the water would all be blown out of the "seals" and the gas would escape as fast as it flowed in.

It is, of course, familiar to everyone that the rate of gas consumption varies throughout the entire 24 hours, what is called the "peak" load coming between sundown and midnight, with a smaller peak in the morning. When the peak is on, the consumption is, of course, several times as great as, for instance, in the small hours when the day is young, and a pipe transmission which would be sufficient if it could be continued uniformly all day and all night is altogether unable to maintain the supply when the demand is greatest.

It is said, therefore, and this is the special excuse for the added monstrosites of recent years, that we must have the big gas-holders to take care of the peak load. Certainly, if we retain both the low-pressure transmission and the low-pressure distribution. With the 2 or 3 in. of water pressure the gas cannot be rushed through the pipes. With a pressure increased to only 15 lb. to the square inch the volume of the gas would be reduced one-half, and it could be driven along at more than four times the present speed, so that pipes of the same size as now in use would transmit eight times the quantity of gas, or as much in three hours as can now be sent through in the 24 hours. This surely would be at a speed sufficient to take care of the peak load, and supply all consumers at all times without the waiting in gas-holders by the way. In this way we have at once a suggestion for the beginning of reform by the warrant it gives for first of all insisting that no additional gas-holders shall be erected anywhere for taking care of peak loads. We have already a long list of locations where gas is transmitted at high pressures to reinforce existing low-pressure storage systems and avoid the necessity of increased holder capacity.

A Mr. Jones, before the Pacific Gas Association, is thus reported: "One of the ambitions of my life is about to be realized in the construction of a steel bracelet around the city of San Francisco for feeding the low-pressure system. This main is now in the ground and is 16 in. in diameter and 7 1/2 miles long. It extends from the old Portrero Gas Works around the city to the old plant we call the North Beach Station. The line is not yet in use for conveying gas, on account of construction work now going on, but it has been under 60 lb. pressure for over 30 days, and has maintained a constant pressure at uniform temperatures both day and night." The piping was entirely successful for the purpose intended, and the preliminary test gave full assurance that there would be no leakage.

What we are certainly coming to is the entire abolition of the hundred-year-old gas pressures, with the gas-holders which cannot survive them, and the service of gas at so-called high pressures—although they would not be high as compared with steam- and compressed-air pressures—directly to every consumer. The following from the *Gas World* (Feb. 4, 1911), an English publication, is reprinted with approval by the *Progressive Age* (March 1, 1911), an able representative of American gas interests. As will be noticed, it goes far beyond the suggestions of the present writer. The article referred to says:

"The introduction of high-pressure gas, when thoroughly understood, will do more for the industry than ever the incandescent mantle did. Its potentialities—its far-reaching utilities—are beyond all power of description.

"All great changes take place gradually, and it is not to be expected that the change from low- to high-pressure gas will be any exception to the rule. Engineers will not jump from 2 in. of water to 200 lb. to the square inch, and yet this is the jump which modern improvements enable any man to take who seriously looks into the question and who realizes what is at his disposal to carry it out.

"With regard to experience, we have at our disposal the record of railway carriage lighting by compressed gas up to 150 lb. or more. In America gas has been distributed at 200 lb. In this country (England) gas has already been distributed to 100 lb., and several miles of mains will be in actual use before many weeks."

The article quoted then goes on to consider the different distribution of costs under the new system, which we need not go into here. Although the high pressures it refers to are all matters of actual record, and in natural gas transmission the pressures go much higher, it would be sufficient for our present purpose to have only 15 lb. per square inch as a maximum working pressure. This would surely render the gas-holders worthless, and if sufficient pressure were put upon the outside of them by the awakened public they would collapse and disappear, property values would reassert themselves over the desolated city areas, and there would be renewed hope for other reforms to follow.

The gas-holder, it may be suggested, is, in a way, like our bad spelling, as some call it; our bizarre weights and measures, as the metricists insist; our Fahrenheit thermometer; our decimal, instead of duodecimal, notation; a thing which started wrong, but which has now become so established that change is not to be thought of. In this case a change insists upon being thought of.

It is not necessary to remind anyone that no gas-holders of the gravity-pressure type are used, or could be used, in the distribution of natural gas, so that they cannot be imperative for artificial gas. As we have seen, they at once become impossible with any increase of pressure; yet gas consumers are requiring higher pressures. The obsolescent fish-tail burner was satisfied with a pressure of 1 1/2 in. of water; the incandescent mantle gives much more light for gas consumed, but it demands higher pressures. Higher pressures are called for where gas is used for heating purposes and much higher pressures are required for gas engines. Gas should be brought to each consumer at a pressure high enough to require a regulator, and this could be individually adjusted to any pressure required, so that everyone could be using it at its best, according to the use to which it was applied.

Object-lesson in High-pressure Gas Distribution.—Instances are now becoming numerous of the distribution of gas, and of its delivery to consumers at pressures above those which are possible with the familiar district gas-holder, so that it is necessarily discarded. Here, for instance, is some account of a recent installation of the St. Louis County Gas Company, St. Louis, Missouri.

There are gas holders of the familiar type at the gas generating plant, where the pressure maintained is from 5 in. to 9 in. of water, the latter being the maximum pressure reached when the tank is full. These tanks are a convenience and a factor of economy, possibly they are a necessity, and in either case if properly located there can be no objection to them, but no other gas-holders are employed for the entire distribution system.

The compressors take the gas at this gas-works pressure and compress it to a maximum of 40 lb. gage. The pipes into which the compressed gas is delivered have a capacity of 20,000 cu. ft., and these constitute the entire storage for the gas after leaving the compressor. As a pressure of 10 lb. is sufficient for all purposes, the permissible range of pressures in this pipe system, about 2 atmospheres, allows fluctuations in the quantity of gas contained of about 40,000 cu. ft. of low-pressure gas, although neither limit of pressure is actually reached in practice. The gas comes to the consumer at whatever may be the pressure in the pipes at the time, and then it passes through an individual pressure reducer, after which it is metered at the constant low pressure maintained.

This is not to be considered as in any respect an experiment, at least with this company, as they have been following this system of distribution for some years, and find that it not only gives satisfaction to all, but that it pays well. They have never experienced any trouble from deposits of any kind in the pipes and the gas is found to be practically as rich after compression and transmission as before. There also has been found no trouble or danger from the heating of the gas in the compressing operation, and there has been no accident of any kind.

In Fig. 64 we see the interior of the compressor room. In one corner of the room, but not seen in this view, there is what is apparently a vertical receiver such as the familiar accompaniment of the air compressor, but in this case it is, instead, a tar extractor through which the gas passes *before* entering the compressors. These machines are two duplex gas-compressors with cross-compound steam-cylinders 12 in. and 23 in. in diameter, and duplex tandem gas-cylinders 17 1/4 in. in diameter with a common stroke of 18 in. and a normal speed of 120 r.p.m. The gas-cylinders are, of course, completely water-jacketed. The mean horse-power is about 120 and the gas-compressing capacity, with liberal allowances, about 1,500,000 cu. ft. per 24 hours. This approximates the present producing capacity, but considerably exceeds the consumption.

The compression of the gas under the conditions here presented is a very simple, or as we might say, a very comfortable job for the compressors. The piston inlet furnishes an ideal means for connecting the intake, and the only automatic control required is a speed regulator, as on a stationary engine. This can easily be adjusted for different speeds according to the rate of gas consumption.

The gas is delivered into an 8-in. main, from which there are branches or continuations of 6-in., 4-in., 2-in., and 1 1/2-in. pipes, the aggregate length of which may be inferred from the

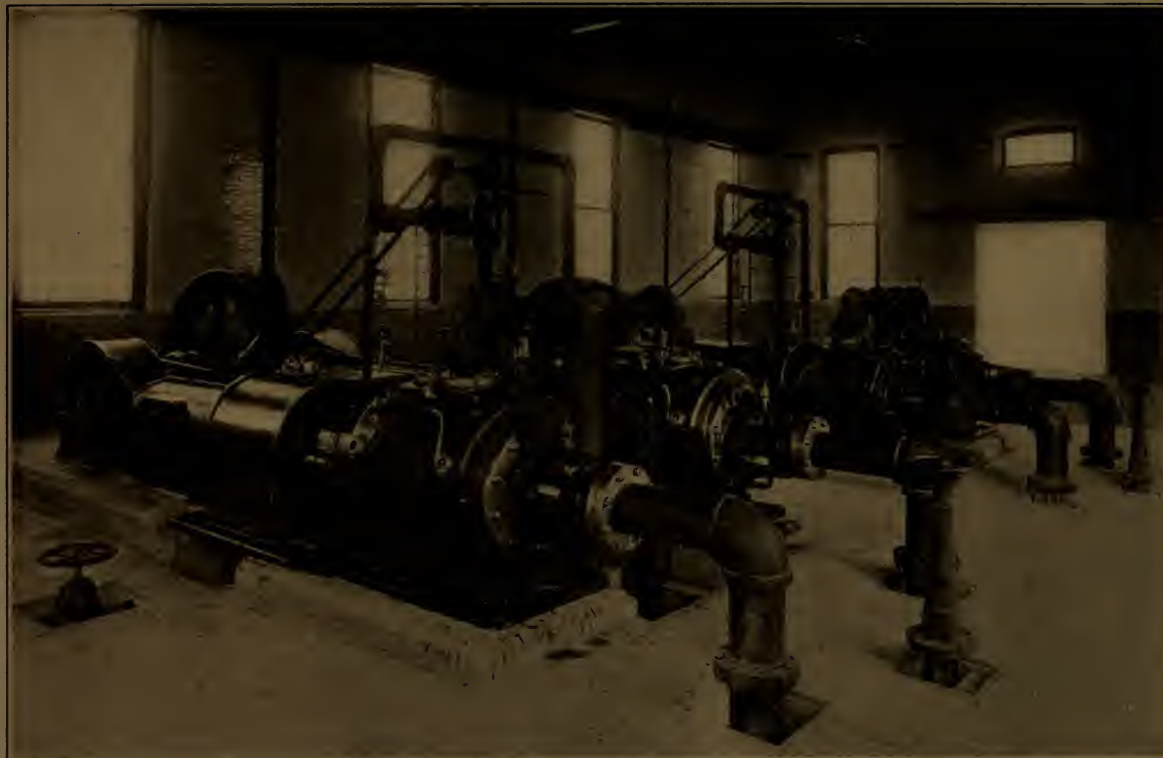

Fig. 64.—Compressor for High-pressure Gas Transmission at St. Louis.

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fact that the area served is about 120 square miles. The present number of customers is 6000, which number is being increased as fast as the pipes can be laid, the present daily output of gas being approximately 500,000 cu. ft. Only one of the two compressors is as yet required, and this is run 16 hours a day.

Fig. 65 is an accurate reproduction of an actual official 24-hour record disk from the recording pressure gage located near the compressors at the beginning of the 8-in. pipe line. This record reads from 8 A. M., Nov. 25, 1911, to 8 A. M., Nov. 26. The

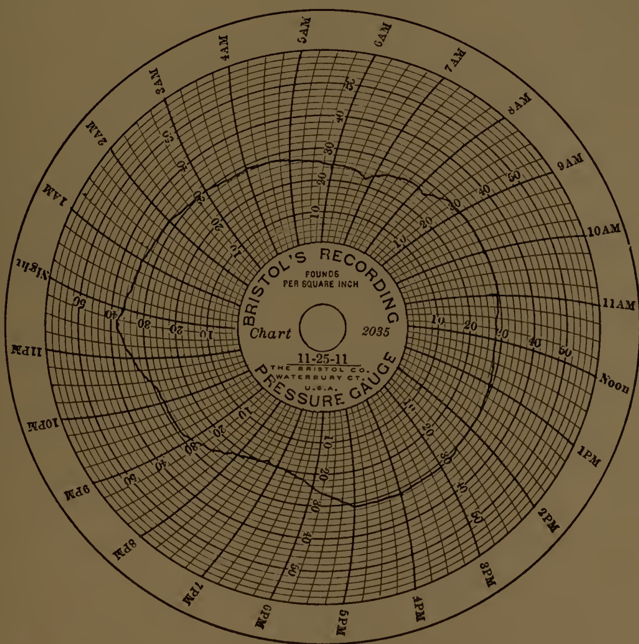

Fig. 65.—High-pressure Gas Record.

pressure line on the record card coincides almost exactly with that of another gage 2 1/4 miles away at the other end of the 8-in. line, so that it is not necessary to reproduce the latter. The combined testimony of the two disks assures us how little loss of pressure there is in this transmission of over two miles, and suggests that the same pipe may easily transmit twice or three times the present volume of gas.

The record line of the pressure-gage card tells its story very clearly. The record begins a little after 8 A. M., and at 9 A.M.

everything is running smoothly, the output of the compressor evidently keeping pace very closely with the eight mid-day hours to 5 P. M., a slight loss of pressure appearing about noon, when we may assume that some extra gas is used for cooking purposes. At 5 P. M. the demand for both lighting and cooking causes the pressure to fall quite rapidly for the next two hours. This is when the "peak" load occurs, the peak being represented on the diagram by a depression.

Before 7:30 P. M. the compressor output has caught up with the consumption. Then the pressure rises gradually until 9:15 and is nearly stationary until 10:15, when there is a rapid rise until 11:30. The upper working limit is nearly reached here and the compressor is stopped, as indicated by the sharp angle at 11:35. From this point until 6:50 A. M. the compressed gas in the pipes is sufficient to supply the demand. The drop is very rapid after 6 A. M. but after the compressor is started the line rises easily before 9 A. M. to the normal day working pressure of about 30 lb. The pressure where the record ends at 8 A. M. seems to be about 2 lb. higher than on the preceding day; perfect coincidence, of course, was not to be expected.

The system of high-pressure gas distribution is constantly extending, both in this country and in Europe, and entirely upon a business basis. It is found that the cost for installation, operation and maintenance of compressor and high-pressure piping is less than that of the much larger low-pressure pipes, the district gas-holders and the land actually required, to say nothing of the depreciated land values which the community has to stand in the vicinity of gas-holders.

The plant here spoken of is a comparatively small one. That the principle it embodies is practically applicable to much larger service is self-evident. That it is not applicable and that it will not eventually be applied to all gas service, however vast or concentrated, it is not safe to assert.

Compressing Natural Gas.—It is not easy to realize the enormous aggregate capacity of compressors employed in natural gas transmission service. Not only are the total volumes very large but the pressures dealt with also are much above those with which we are familiar in ordinary compressed air practice. Natural gas is now produced in many localities and the transmission extends to great distances in many if not most of the gas districts.

The compressor would seem to have a claim upon every gas well for ultimate employment. The pressures are generally high to begin with, and then the gas can laugh at the compressor if it has not too far to go; but as the pressure invariably falls off there comes a time when the compressor must be called on to boost. Then the distances to which the gas must be transmitted are often so great that no matter what may be the initial pressure the help of the compressor is needed before the gas can be relied upon to reach the most distant consumer.

A typical case of commercial gas transmission of considerable proportions is that of the City of Cincinnati, Ohio, and the surrounding towns, the brief sketch of which here given being made up from information furnished by Wm. A. Miller, Gen Mgr. Gas Dept., Union Gas and Electric Co., Cincinnati.

The pipe line from Culloden, W. Va., to Cincinnati, Ohio, consists, tracing it backward, of 123 miles of 20-in. seamless steel pipe, from Cincinnati to the Big Sandy River compressing station, and 33 miles of 18-in pipe from the Big Sandy to the end—or beginning—of the main line near Culloden. From the 18-in. main pipe line there are about 40 miles of lateral 12- and 8-in. pipe lines extending to and penetrating the gas fields which embrace about 300,000 acres, on which are drilled about 100 gas wells having an open flow of approximately 200,000,000 cu. ft per day.

The West Virginia pipe line, used in connection with storage holders (capacity, 10,000,000 cu. ft.) is capable of transmitting to the consumers upward of 75,000,000 cu. ft. of gas daily, with an initial pressure of 320 lb. at compressor station, and a terminal pressure of about 60 lb. at Cincinnati.

The distribution system of the City of Cincinnati and villages adjoining, consists of about 45 miles of belt line in the form of a figure 8, outlining the whole territory to be supplied, upon which a pressure of from 3 to 5 lb. is maintained, varying with the demand for the gas and the temperature of the atmosphere.

From the belt line the gas is caused to pass through forty-two district regulators set in masonry pits, and located in streets in different parts of the city and villages supplied, which regulators are adjusted for maintaining a pressure of 5 oz. on their outlets. The outlets of the regulators are connected to the low-pressure system of mains, consisting of about 650 miles of all sizes of main pipes, from 4 to 30 in. in diameter. Individual house regulators are only used where the belt line is the only main from which a supply of gas can be obtained for the customer to be supplied.

Information somewhat more detailed as to natural gas compressor practice is here abstracted from a paper in the proceedings of the Engineers' Society of Western Pennsylvania by Mr. E. D. Leland.

It is stated that in 1892, at Greentown, Indiana, there was completed the first station designed for compressing large quantities of natural gas to extremely high pressures. The problem was to continuously deliver at Chicago an adequate supply through two 8-in. lines 120 miles long. The compressors being installed a considerable time before the pipe lines were ready, they were first used to compress the gas into large steel tanks under a pressure of 700 lb., and these tanks were shipped to Chicago.

The pipe lines when completed were tested at 600 lb. air-pressure, and the entire station was designed for these high pressures. The ordinary delivery pressure was somewhat lower, but the compressors proved amply able to compress the gas up to the maximum pressure whenever required.

As storage capacity for a sufficient reserve in or near Chicago was entirely out of the question, it was most essential that a continuous delivery should be maintained by the pipe lines and the compressing station alone. Hence the installation consisted of twelve straight-line, single-stage, steam-driven compressor units of moderate size, so that an accident to any one machine would not seriously affect the delivery capacity of the station. Later in the history of gas compressing larger and fewer units were used. As early as 1896, the Fort Wayne Gas Company installed some large cross-compound Corliss engines which were used until the practical exhaustion of the Indiana gas field, and these were then sold for use in the Ohio and Kansas gas fields where they give good service.

Single-stage compression was satisfactory for the twelve machines spoken of above as long as the high intake pressure was maintained; but when the pressure fell off two-stage compression was resorted to, using eight of the machines for the first stage and the other four for the high pressure and delivery work.

A gas compressor is practically an air-compressor designed to operate under the high pressures usually required in natural gas transportation, and no experimenting is now required in order to obtain reliable compressors, nor can there be any valid excuse for failure to deliver gas on account of defective machinery. An instance is cited of four compressors installed in 1904 which have been in almost constant operation ever since. They have been steadily compressing gas to pressures ranging from 200 to 275 lb., and are still operating successfully with the original intake and delivery valves that came with the machines.

In some cases the high-speed gas engine, connected by rope drive to a slower speed and longer stroke compressor, proves a good arrangement. It not only affords relief from the undesirable features of the short-stroke compressor, but it also makes it feasible to locate the engine at a safe distance from the compressor. While the necessarily quick starting of a gas engine is more liable to cause trouble, by breaking the ropes or by throwing them out of their grooves, than is the case with the slower starting steam engine, still with this type of plant fairly good results have been obtained.

For continuous and reliable service in compressing large quantities of gas, the modern cross-compound Corliss engine, directly connected to the compressing cylinders, makes an ideal installation. This engine is particularly well adapted for the long stroke and moderate speed, so engineers thoroughly understand that the changing pressure conditions in field and main lines are also well met by the flexibility and high overload capacity of this type of prime mover. So well is this fact recognized that in the stations delivering gas to the Pittsburgh district we find 39 Corliss engine driven compressors installed, comprising a total maximum capacity of over 76,000 h.p.

CHAPTER XX

GASOLINE BY COMPRESSION—LIQUEFIED NATURAL GAS

The present chapter has to do with an industry which has been developing rapidly, and the details of which have been becoming matters of general knowledge, while the preparation of this book has been in progress, and it cannot expect to be entirely up to date while developments and revelations are still occurring. The demands of the automobile and the auto-truck have advanced the value of gasoline enormously, and have stimulated the search for new sources of supply. Under this impulse there has been growing a new industry which is understood to be proving highly remunerative and to have the merit of conserving one item of our natural resources by producing value and usefulness from one of the waste and hitherto unconsidered effluents of the oil industry.

The general public has understood that there are oil wells and gas wells, but in fact many wells are, and most wells have been or may become at some time, either or both of these. So also the public has known of in a general way, and has deprecated, the enormous wastes of both oil and gas from their respective wells through the failure to provide the means of conveying or of storing, and thus of ultimately utilizing all of either product which has flowed forth too copiously when the wells have been first opened, or of choking off the flow until arrangements could be made for its economical disposal. Both individual or corporate enterprise and legal enactments have made such waste less frequent now. It often happens, however, that when the gas comes as the accompanying product of a producing oil well, or when the gas continues to flow after the pumping of oil has ceased such gas all goes to waste because it is thought that it will "cost more than it comes to" to save it. It appears now that there is money in it, and also that money is being got out of it. The following account is mostly obtained, condensed and rearranged, from a full and clearly written description in *Mines and Minerals.*

Most oil wells at the beginning are self-flowing, or are even gushers, yielding five or six hundred barrels a day, and in California much more than that, but the decrease in production is rapid, they become pumpers and then are pumped only at intervals until they finally yield less than a barrel a day. The opening of new wells goes on continually and the number of abandoned wells is very great and always increasing. All the time, and long after the yield of oil has ceased there is a flow of gas from the wells, and it is this gas which in later years has made most of the waste.

The beginning of the recovered gas industry began really with the gas piped and thus saved from oil wells rather than from the entirely abandoned gas. In the long pipe lines it has been a common occurrence for gasoline to collect wherever a down bend in the pipe has made a pocket. Water also is deposited in the pipes and the re-evaporation of the gasoline through leakage or otherwise may so reduce temperatures as to cause annoyance and serious trouble by freezing the water and choking the pipes, and in seeking a remedy the process for manufacturing gasoline from the gas has developed and since its perfecting on practical lines it is being applied to the abandoned gas-yielding oil wells above spoken of. The plants now employed, numerous and not individually very large, are equipped with a refinement of appartus and method which has been perfected after considerable experimenting and selection.

The cut, Fig. 66, shows the layout of a typical West Virginia gasoline plant with a capacity for treating 150,000 cu. ft. of gas in 24 hours and producing 500 to 800 gal. of gasoline at 92° Baumé. The gasoline is shipped in 50 gal. steel barrels which are hauled by wagon to the nearest railroad station, and the gas also no longer flows to waste but is piped to the nearest natural gas pipe line, usually not distant.

The gas is compressed in two stages, or successively by two 35-h.p. gas engine driven, straight-line air-compressors. The first compressor which may have a cylinder from 6 to 12 in. in diameter draws the gas from the piping system connecting all the available wells in its immediate neighborhood, and from an intake down to, say, 15 in. of vacuum the gas is compressed to 20 or 30 lb. It then passes through a water cooler to the second compressor with a cylinder diameter of, say, 4 1/2 in., where it is compressed to 150 lb. or over. This final pressure must be determined by trial, as the process depends considerably upon

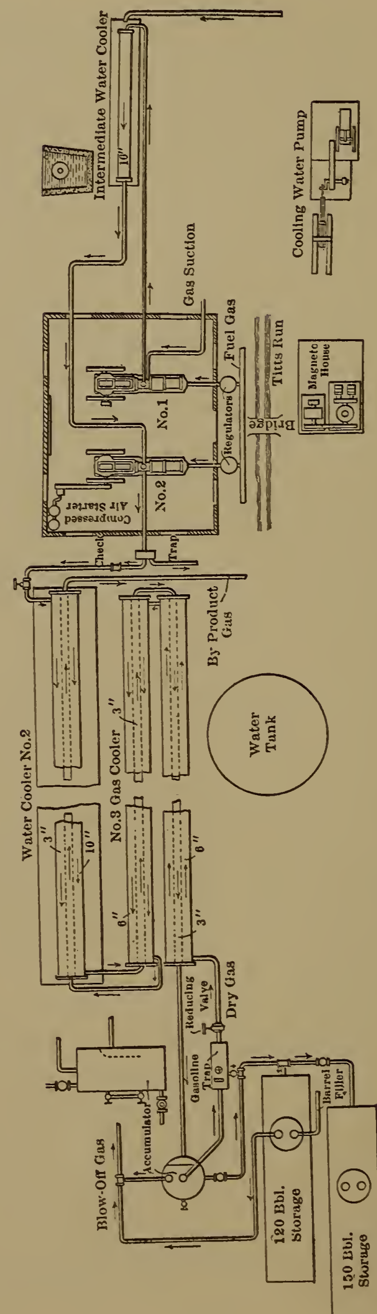

FIG. 66.—Layout of Gasoline Plant.

the quality of the gas and the refrigerating treatment. In some plants more gasoline is produced with a terminal pressure of 150 lb. than with 250 lb.

The gas at 150-lb. pressure passes through an 80-ft. water cooler, and then through a double 80-ft. gas cooler, the latter using the by-product gas for cooling. The saturated refrigerated gas under a pressure of 150 lb. or over, enters the side of the accumulator tank at a point about two-thirds its height, measured from the bottom. A baffle plate riveted in the tank deflects the flow and precipitates the gasoline. The accumulation of gasoline is shown by a gage glass, and periodically the attendant blows it into a storage tank of 120-barrels capacity. The storage supply stands at about 20 lb pressure. From the stock tanks the gasoline is loaded into the steel barrels.

The crank case on some of these engines is closed and has a vent pipe leading above the building; thus, any gas leaking from the cylinders will be carried out of the building and the danger from fire or explosions is lessened. The make-and-break spark system is used for ignition, and a friction-driven magneto for each engine is located in a small building some 100 ft. distant. A small gas engine operates these magnetos and also the generator for lighting the plant. An air starting outfit, consisting of one air pump compressing to 150 lb., air-receiver, starting valves, pressure gages, etc., makes the starting of these large gas engines an easy and a safe operation. About 2 lb. gas pressure is used for the gas-engine service. A regulator placed outside the building is necessary to deliver the fuel at a constant pressure.

The Cooling System.—All the gas-engine and compressor cylinders are water-cooled. The gas as it comes from the wells may be at about 60°. The heated gas from the first compression goes to a water cooler consisting of a concrete vat $20 \times 4 \times 4$ ft. A continuous flow of cool water passes through this tank in which a 10-in. pipe is laid lengthwise along the middle and to which the 3-in. delivery pipe from the first compressor is attached at one end, while to the other end an inlet pipe to the second compressor is attached.

The high-pressure gas from compressor No. 2 passes first through an automatic separator which removes any lubricant that may be carried over from the compressor. It also catches any gasoline that may drain back from the second cooler. This second cooler is 80 ft. long and consists of a concrete tank like the intermediate cooler. A 10-in. pipe is placed lengthwise of the tank. The 2-in. delivery pipe enters a special fitting at the rear end of the 10-in. pipe, and the water-cooled gas passes out by a 3-in. pipe 80 ft. distant. This 3-in pipe enters the end of an 80-ft. length of 6-in. pipe, and returns through a second 80-ft. length of 6-in. pipe and thence to the accumulator tank. From the gas space in the top of the accumulator tank a pipe leads to a gasoline trap which collects any gasoline that may be carried over from the accumulator and returns it to the stock tank. This trap is also fitted with a pop safety valve that relieves the accumulator from any over pressure and delivers the gas that may be blown off to the fuel-supply gas mains. Above the gasoline trap the by-product gas passes through a reducing valve and enters at low pressure the lower 80-ft. branch of the 6-in. gas cooler. This No. 3 cooler is made up of a loop of 6-in. pipe laid in a box packed with sawdust.

The peculiar design of the cooling system makes it necessary to use some specially designed pipe fittings. The cooling effect of the expanded by-product gas is considerable, as it flows 160 ft. through the 6-in. pipe that encloses the 160 ft. of 3-in. pipe carrying the compressed gas to the accumulator. From the 6-in gas cooler, the expanded gas goes to the 80-ft. water cooler and passes through a 3-in. pipe laid lengthwise through the center of the 10-in. pipe. From cooler No. 2 the by-product gas goes through a 2-in. pipe to the gas engine feed-line. The by-product gas not used by the plant goes into the natural gas mains and is sold. The overflow water from the concrete tanks flows by gravity to the water-jackets of the engines and compressors. The by-product gas is a blue-flame gas that is more desirable for fuel and lighting than the raw gas, as it does not deposit any soot or blacken at all the furnaces, gas mantels, cooking utensils, etc.

The partial vacuum produced by the first compressor as it draws its gas supply from the wells aids both the oil and the gas production; in fact, in some cases the gas is given to the gasoline plants, by the oil man, as the increase of oil due to the vacuum is quite an item to the well owner. At the same time, however, the owner draws back all the by-product gas he needs for pumping the oil.

The vacuum in the field lines is one of first importance in gasoline production. So great is this feature that in some gasoline plants a special independent low-stage vacuum pump and gas compressor has been installed so as to regulate the field pressure and increase the production. With this low-pressure compressor 24 in. or more of vacuum can be held on the wells, and at the same time the efficiency of the regular compressor units not be lowered. The advantages observed from the use of the vacuum system have led to the reopening of abandoned oil fields, not for the oil but for the gas from which they make gasoline.

The business of making gasoline from natural gas is necessarily a hazardous one. It is a new business, and this, coupled with the usual combinations of ignorance and carelessness, makes a list of accidents that one would naturally expect. Even the empty barrels have exploded when standing exposed to the hot sun and with the vent plugs set tight. On one occasion an empty went up when standing on the freight station platform. Those old in oil-well service have become accustomed to handling nitroglycerine, and while they respect it, they treat it with a feeling of contempt. On the other hand, gasoline of from 92° to 100° Baumé is a new thing to them, and they have got "burned" as a consequence. Therefore, to hear an operator about a gasoline plant remark that he would rather carry "nitro" than gasoline is evidence of the fear in which it is held.

Portable Liquid Gas.—The preceding may be said to merely outline what may be considered the crude beginnings of an industry which under cooperative scientific investigation and ingenious application of the principles discovered has resulted in a successful method of liquefying natural gas and transporting it for use, especially in isolated localities. Dr. Walter O. Snelling, who has been a leader both in the experiments and investigations preceding and in the applications resulting, gives a summary of the results of the new processes in a paper before the Pittsburgh Section of the American Chemical Society. Tests made with great care, he says, have shown that it is possible to produce from natural gas, by a combination of the methods of stage compression and rectification, not only gasoline of the most excellent quality, and equal in every respect to the best grades of refinery product, but in addition to recover all of the ethane, propane and butane, as a liquid gas, in such a form as to make it a convenient, safe, clean and cheap method of lighting isolated dwellings, such as country homes, seaside resorts, lighthouses and light buoys, etc.

The name "Gasol" has been given to the liquid gas produced by the new process. When under a pressure of 500 lb. to the square inch it exists as a clear, transparent liquid. When the pressure is lessened it changes into gas, and at normal or atmospheric pressure this gas is, unlike the gas produced by blowing air through gasoline, for example, extremely dry, and does not give any liquid condensate in the pipes, etc., all of the hydrocarbons which so condense having been separated in the process of rectification and going into the gasoline product. One volume of liquid Gasol produces about 250 volumes of gas upon release of pressure, and the gas so produced has a heating value of about 22,000 calories per liter, or about 2400 B.t.u. per cubic foot. When it is remembered that the heating value of ordinary coal gas is only about 600 B.t.u. per cubic foot, and manufactured oil gas is less than 650 B.t.u. per cubic foot, it will be seen that the new gas has about four times the heat-producing capacity, when equal volumes are considered, of either coal gas or manufactured oil gas. In addition, its flame temperature is much higher, being decidedly higher than the flame temperature of natural gas or any other of the common gases used for heating. The flame temperature of ordinary natural gas burning in air is about 2150°, and the flame temperature of ethane burning in air is about 2205°. The flame temperature of the new gas is about 2300°, and since the amount of light produced from the Welsbach mantle bears an important relation to the temperature of the flame, the reason is here seen for the remarkable brilliancy of the light produced by the new gas, which excels in this respect all gases previously known.

The new process of preparing a liquid gas of perfectly homogeneous nature and uniform composition is the culmination of the work of many men, and of several years of experimenting. It opens to the use of the world the enormous volumes of oil-well gas now so generally wasted, and produces from this waste material a product which gives to the country home all of the convenience of gas for lighting and cooking, thus giving to the farm advantages which have been up to now available in general only to the city.

FIG. 67.—Six Drills in the Heading–Lookout Mountain Tunnel.

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CHAPTER XXI

ROCK-DRILL DEVELOPMENTS

It would seem that nothing could be more deserving of honorable mention in these pages than the rock drill. While it is so entirely dependent upon the air-compressor for the maintenance of the most responsible and effective of its activities, nothing has done so much retroactively to stimulate the development of the air-compressor or to promote the growth of compressed-air apparatus in general.

If we were to ask one only casually acquainted with the rock drill to tell us what it is and what it does, we would perhaps learn from him that it is one of the simplest of machines, its sole function being to reciprocate a piston with considerable rapidity, the piston carrying a steel bit which by constant forcible striking of the rock at the same spot gradually crumbles its way into it, forming a hole of any required depth. This would not be an untruthful statement, except as to the first item, but it would be an absurdly inadequate one. The simple fact is that there are few established inventions which so completely satisfy so many imperative and exacting conditions as does the rock drill.

What is Required of the Rock Drill?—The drill, in a way, does rough work, rough surroundings are its habitat, and it is handled and operated by men who would also, perhaps unthinkingly, be characterized as rough. It must therefore in itself be rough and strong exteriorly, so that it can stand the most strenuous use and unlimited abuse. It must be able to put in holes in the rock in any direction, vertically downward or vertically upward or at any angle between, and in any vertical plane all around the circle.

It must be set in the required position quickly and precisely, and while drilling each individual hole it must be held accurately and securely, yet never rigidly, in position and direction, and provision must be made to constantly feed forward the drill as the steel advances into the rock and to withdraw it for changing bits or for starting another hole. This suggests the requirements for the drill mounting, which are entirely separate from those of the drill itself and are trivial in comparison.

Perhaps the most important and responsible detail of the drill is its valve and valve motion. There can be no positive valve movement as with a steam-engine, for the double reason that the stroke of the piston in the drill cylinder constantly varies and that if an otherwise satisfying mechanical movement and connections could be devised the mechanism would be soon demoralized by the jar of the machine; and yet the valve must and does admit the air or steam freely and promptly so that the piston moves with the required force; the piston must normally avoid striking the cylinder heads, and whatever the position of the drill or of the piston in the drill cylinder it must always and instantly obey the throttle. The possibility of occasionally striking the cylinder heads seeming to be unavoidable; they cannot be rigidly secured to the cylinder but must yield a little upon occasion for their own salvation.

The drill bits, of course, wear in use, and in hard or gritty rock the wear may be rapid, so that it must always be possible to change a steel for another; the chuck must therefore hold the bit securely under the shock of the repeated blows, and yet it must be tightened or loosened without delay. This constant changing of bits, and the minute movements of the shank in the chuck under the repeated blows, of course entails wear, which must be provided for, and the chuck or its interior must be renewable so that the rectilinear truth of the successive bits when inserted may be relied on.

Constantly while the steel is advancing into the rock it must be kept in rotation, the turning of piston and steel not having any regularity of angular movement, but still insuring that the cutting edges of the bit shall each time strike in a different place. The automatic rotation provided, while it insists upon the turning of the piston and bit more or less for each stroke, still does it in a yielding way, so that accidents or breakages do not occur if at any time the bit refuses to be immediately turned as far as the rotation device would normally require.

It was conceded at the beginning that the rock drill is in a way a rough machine, and yet in the details of it there is scarcely any machine so carefully made and so precisely sized in all responsible dimensions. The drill, for instance, is, as we know, normally driven by compressed air, and the readiness of compressed air to leak away wherever it finds the chance is more than sufficiently notorious, and yet the cylinder heads, the valve chests and other parts requiring air-tight joints are so perfectly surfaced and fitted that they go together, and are always used, metal to metal, absolutely without packing.

The rock drill leads the strenuous life. It is necessarily subject to severe wear, much abuse and frequent accidents, so that its parts are liable to require renewal or replacement at any time. Every part of every standard drill is made so accurately to gages that stocks of such parts are maintained wherever the drills may be employed in any part of the world, and these pieces interchange with and replace each other as though there were but one machine of its type in existence.

The manufacture of the rock drill as it is to-day imperatively requires the coincident compliance with many exacting conditions. After the drill has come to be practically adapted to its special line of work by the successive accession of innumerable inventions, the trying out of the successful details, the determining of the absolute and the relative sizes of the several parts for their mutual cooperation, the selection of the most suitable and enduring material for each part and its proper heat treatment and manipulation to secure strength and slow wearing qualities, the system and means of production are to be elaborated. This means carefully collected and trained and long experienced workmen, installations of automatic and often special machinery, costly collections of gages, jigs, templets, drills, taps, reamers, of original tools for special operations, and the breaking in, the testing and the rigid inspection of the product before it is sent out.

The rock drill began its life work in tunnel driving, and suitable mountings were provided for it. Now it is used in sizes larger and smaller for drilling both below and upon the surface, and it is held indiscriminately upon tripod, shaft bar or column, upon the sliding and angularly adjustable frame of the gadder and in the heavier sizes for subaqueous surface drilling; and it is further transformed in the self-contained channeler and the coal-puncher.

These suggest the lines of growth and development of the original percussion drill with reciprocating, bit-carrying piston. It may be said to be some 60 years old, but in the last score of years, even in the last decade, its status and prospects have been rapidly changing.

At the very beginning it was a radical departure in principle of operation from the hand-drilling process which it was to supersede. In the latter the drill was held stationary against the rock with usually two men striking with sledges upon the head of it, there being some hand rotation of the drill after each successive blow. The new drill seems to have been such a success that no one for the time thought of questioning the correctness or finality of its working principle.

Then came along the "pneumatic tools," so-called, designed to take the principal part thereafter in the former exclusively hand operations of stone carving, the chipping, caulking, riveting of the metal worker, and work of that class in all the trades. These tools returned at once to the principle of operation which had lived through the ages in hand work, the chisel, or whatever the tool might be, being pressed upon the work and the blow being showered upon the tool socket. The "pneumatic tools" were such a pronounced success from the start, and have so rapidly developed their adaptability and extended the scope of their actual employment that their incursion into the rock-drilling field was inevitable, and accordingly we have already lines of rock drills actuated upon the pneumatic hammer principle and with pronounced success. These drills have not so much crowded the reciprocating piston drills in their legitimate lines of work as they have struck out new lines for themselves where the drill of the old type was at a disadvantage.

What immediately follows I mostly abstract from an editorial in the *Engineering and Mining Journal*, the writer of which is evidently more closely in touch with actual mining practice as it is to-day. Three types of machine drills, he says, are now well recognized: The piston drill, always mounted, with reciprocating drill-bit rigidly attached to the piston-rod, automatically rotated and fed by hand, and with the cuttings removed by gravity or by the churning of the bit in water introduced at the collar; the hammer drill, always mounted, in which the bit does not reciprocate, but is automatically rotated and fed by hand, the cuttings being removed by a stream of air or water through the steel, the piston striking the shank of the steel a blow similar to that delivered by a hand hammer on the head of a nail; and the "jackhammer" or "plugger" drill, always unmounted, in which the bit does not reciprocate but is rotated by the operator and is fed by the weight of the machine and the operator, the cuttings being removed by gravity or by a stream of air or water through the steel. This latter is essentially a one-man tool, held in the hands, progress being made either through the weight of the drill, through the pressure by the operator, or both.

The hammer drill proper is suitable only for drilling upwardly slanting holes, and consequently finds its field of usefulness especially in stoping, whence the name "stoper" often applied to these drills. The jackhammer, on the contrary, cannot very well be used for upward holes on account of the inconvenience to the operator, who holds it without support, but as a tool for drilling down holes it is marvelous. At present it is being employed in shaft sinking at the Newport mine, in Michigan, and we hear with highly economical results. The men take these instruments with them just as they used to take hand hammer and steel in olden days and save the time of rigging and unrigging the heavier machine drills, with obvious advantage. So far as we are aware, the shaft sinking at the Newport mine is the first use of the jackhammer in such work.

It is thus brought out that each of these types of drills has its special field. In drifting, the piston and the hammer drill divide the field, the two-man or the one-man size being required according to the difficulty of the ground. In sinking, the piston drill is still pre-eminent, but the hammer drill, and especially the jackhammer, have been rapidly winning favor even for steep down holes, and the jackhammer is being successfully used also in shaft sinking. In "raising" the hammer drill or "stoper" is at its best, but in work rising at a low angle the piston drill is still supreme. In stoping the choice is determined by the nature of the deposit. If the deposit is flat or if it is taken off in flat benches, then the piston drill is the logical choice. If it is steep, comparatively narrow and mined by overhead stoping, the hammer drill is much superior. If glory-hole methods can be adopted the jackhammer will probably be selected.

In some of the Michigan copper mines the small drills have contributed to remarkable results. In these the piston drill of the old type has been gradually replaced by a newer and later type of piston drill which is partly, if not entirely, a one-man machine. Recent figures in bench or footwall work have shown that the cost of mining with this light piston drill is about $0.56 per ton, each machine giving about 11 1/2 tons per man per shift. With the hammer drill the cost per ton in the same, or similar, material is reduced to about $0.45, each man producing about 12 1/2 tons per shift.

In work where the jackhammer can be used to advantage it is obvious that a still greater efficiency ought to be obtained, because of the saving of time in carrying and mounting the drill. In fact, increased efficiency is actually realized, and to a high degree, the cost per ton with the jackhammers being about $0.15, each man producing about 36 tons per shift.

In these comparative figures of cost no overhead charges are included nor the cost of the air. The former will be different at each mine and with each organization and the cost of compressed air also is variable, so that comparisons may be made more fairly without including these. In a general way it may be assumed that it costs from $0.50 to $1.00 for 24 hours per drill for compressed air. With the jackhammers the cost is reduced by approximately one-half as compared with drills of the other types.

The old type of piston drill weighs about 250 lb., the newer and lighter piston drill about 150 lb. and the jackhammer only about 60 lb. Naturally its lighter weight and the usage to which it is subjected result in a shorter life, but its first cost is only a little more than half of that of the piston drill, and its efficiency is so much greater that it is economical to adopt them, wear them out, and replace them with new ones as required.

Some marvelous work in reducing mine costs has been done in the Michigan copper country during recent years, and this has been achieved largely by the introduction of improved methods of drilling and improved drills, in which it must be said that the manufacturers have rendered highly valuable assistance to the mine engineers. In fact in mining machinery as in other things the principal inventions have originated outside of the mines and the inventors and designers have not been miners; although necessarily they have been more or less informed as to mining possibilities and requirements.

The progress in mining drill practice and the increase in general efficiency is after all not to be credited to the drills alone. There have been improvements apart from the drill itself which have directly or indirectly helped to augment the results accomplished by the drill; that is in the records of cubic yards of rock broken or of tons of ore got out.

For one thing, the modern high explosives should have due credit. The enormous advance in the effects produced has made

FIG. 67a.—Typical Stunt with a "Jackhamer" Drill in a Montana Copper Mine.

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it unnecessary to drill so large holes in the rock, and this has helped to provide the opportunity for the light hammer drill, or for the one man drill in general. With the old black powder and the holes which it would have required, both in size and number, to produce the same results the case would have been different.

In Figs. 68 and 69 we have a rather curious illustration of the difference in the force and effect of the old explosives and of the

Fig. 68.—Effect of Black Powder.

Fig. 69.—Effect of Dynamite.

new. The first shows the bottom of a hand drilled hole on the line of the original Croton aqueduct located about half a mile below High Bridge, and the other, not a quarter of a mile from the first, shows the bottom of a machine-drilled hole on the side of a street only recently cut through the rock. In the first case the outline of the hole is still perfect, showing that the powder used had not force enough to shatter the rock at all. In the second case where dynamite, or some otherwise named modern explosive, was used the force was so intense and concentrated that the rock immediately surrounding the charge was so shattered as to leave no trace of the shape of the drilled hole for a couple of feet of what was evidently the lower end of it.

The Drill Sharpener.—But another thing which is helping the drills to accomplish more and to establish new records is in the better maintenance of the shape, size and condition of the cutting ends of the drill bits. The hand dressing of rock drill steels has always been so unsatisfactory that it would seem that the invention of a machine to do the work was inevitable. The lateness of its arrival showed that the devising of such a machine was no trivial task for the inventor, and the perfect success of the latest machine speaks for itself of the inventive skill and the thoroughness of adaptation of means to ends in the work of the designer.

The machine is, first of all, as it necessarily must be, more properly a drill maker than simply a drill sharpener, since it takes the end of the original bar of steel and upsets it and shapes and sizes it for its first work, also forming the shank on the other end, and the simpler job of "sharpening" the steel, or of restoring its shape and size comes later.

The machine is pneumatically operated throughout, using the air pressure provided for running the drills. The end of the steel is heated as hot as is permissible and then, in a horizontal position and regardless of the length, it is placed in the machine and is tightly gripped by a vertical movement of the upper portion of the vise. The jaws which grip the steel also form, when closed, a conical die for shaping the outside of the bit. Immediately that the steel is gripped a "dolly" is advanced and pressed against the end of the hot steel and simultaneously with this movement a pneumatic hammer action showers rapid and forceful blows behind the dolly and the steel is quickly upset to fill the die and the cutting face of the bit is shaped at the same time. The entire action of the machine is controlled by a single lever and the entire operation is one to be watched with pleasure by any one endowed with a modicum of mechanical instinct and sympathy.

The advent of such an efficient machine as this changes the attitude of the drill runner at once. It is no longer necessary for him to be thinking of saving blacksmith work or to continue to use steels after they lose their gage or have ceased to cut at their best. Not only do the steels cut more rapidly when working but they follow each other better and there is less difference in gage between the starter and the bottomer. The more frequent changing of the steels which the mechanical drill sharpener encourages is an important time saver rather than the reverse, and where more than two or three drills are working together the machine should soon pay for itself.

CHAPTER XXII

THE ELECTRIC AIR DRILL

It is generally the deliberate purpose of the present volume not to describe in detail individual devices which have been developed and have found successful employment in compressed-air practice. The electric air drill, however, is in a class by itself, and may claim mention somewhat anomalously from the fact that, while it is an alert and winning rival of apparatus which belongs entirely to the compressed-air class it cannot properly be included, in any precise enumeration of compressed-air devices. It ignores absolutely the standard air compressor and the pipe line. It is in fact also not at all an electric drill, and might as well be driven, if equally convenient, by a belt or by a steam-engine direct. The name which has been given it, intended to be descriptive of it, is not at all satisfactory, and is apt to be misleading without accompanying explanations.

The application of the electric current to the direct and immediate driving of the percussion rock drill has been an increasingly urgent problem for a score of years, but there has been little reason at any time to hope for a satisfactory solution. With the advent of the electric air drill the problem is not indeed solved, but it is annihilated. There is a better way, as this drill shows us, to apply the electric current to rock drilling than by means of any bona fide electric drill. This drill, in its ultimate actuating element is actually an air drill rather than an electric drill, using the electric drive, indeed, for the motor detail of it in the most advantageous way possible, without any of the disadvantages inseparable from the electric drill as such, with all the advantages of the best compressed air driven drill, with special additional valuable features of its own, and without any accompanying disadvantage.

These are large claims, but an understanding of the drill and its mode of operation would seem to justify them. Fig. 70 shows an electric air drill as used in actual work with all the apparatus pertaining to it. The drill itself looks quite like a standard com-

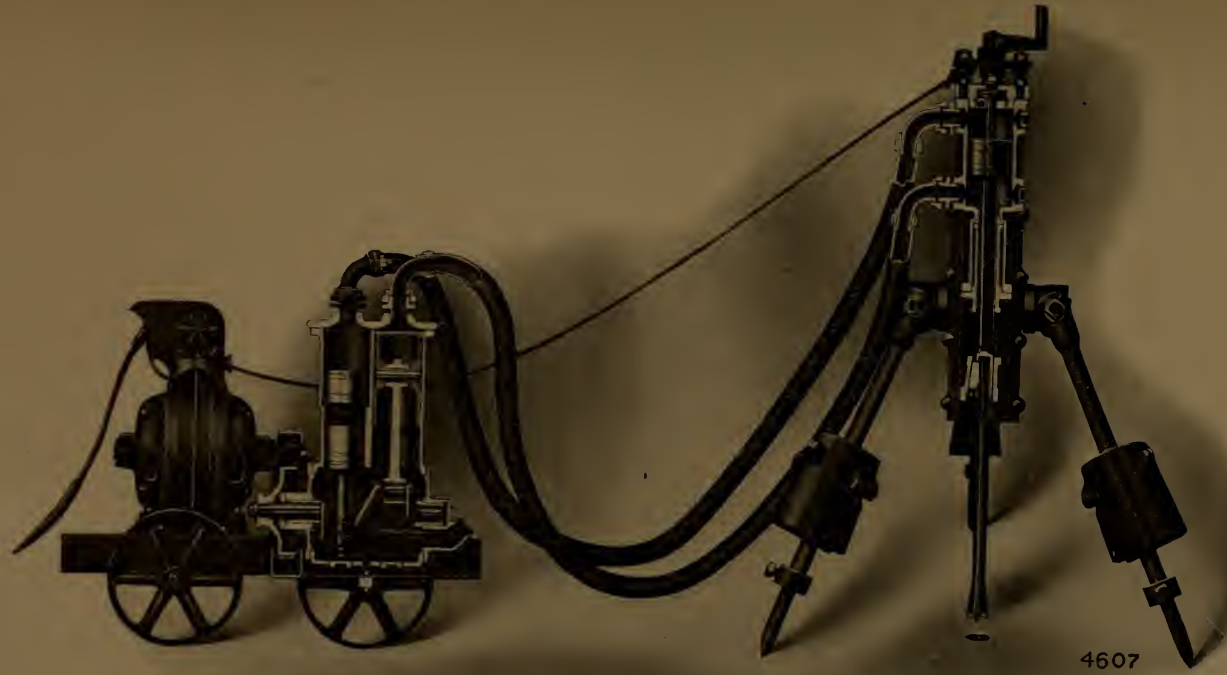

Fig. 70.—Electric Air Drill Complete.

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pressed air driven drill, and it may be mounted upon tripod, bar or column in the usual way. There is the usual drill cylinder and piston, the piston rod carrying the bit or steel which does the actual rock cutting. The cylinder slides in a shell, moved back and forth therein by the usual hand operated feed screw. Near each and of the cylinder there is a projecting boss for attaching a hose, with free, direct openings into the cylinder. The piston is short and the piston rod, which runs in a long sleeve extension of the cylinder, is much larger than in the usual air drill, so that no enlargement of the piston rod for the chuck portion is required. The chuck used, which is an incidental and independent invention, is a special self-tightening device fitting the taper end of the rod.

The little truck, one of which accompanies each drill and is an integral part of the complete apparatus, has a small electric motor, either direct or alternating current, driving by reducing gears a shaft with a crank on each end which cranks give alternating movement to two vertical, single acting trunk pistons in corresponding air-cylinders. One of these cylinders is connected by a short length of hose to one end of the drill cylinder, and the other end of the drill cylinder is connected by a similar hose to the other air-cylinder. The body of air thus enclosed in each air-cylinder, in its hose and in the connected end of the drill cylinder is never discharged or changed, but plays back and forth as the piston moves. The air-cylinders are thus not compressors but pulsators, as their function is not only to furnish effective driving pressure to one side of the drill piston but alternately also to reduce the air resistance at the other side of the piston.

The principle of operation is simple and easily understood. The entire apparatus, both of the hose, both pulsator cylinders and both ends of the drill cylinder, in normal working condition is filled with air at about 30 lb. pressure. The correct supply of air is accumulated and maintained automatically. If through leakage or otherwise the charge of air becomes insufficient there occurs in the cycle of operation a moment when the pressure of the air becomes less than that of the atmosphere, and at once an inwardly-opening check valve admits sufficient air to supply the deficiency. This arrangement also operates to charge the system with sufficient air when first starting up.

The air being then at normal pressure, as the cranks revolve the piston in one of the pulsator cylinders advances and the pressure is increased in the connected end of the drill cylinder. At the same time the pressure in the other air-cylinder and at the other end of the drill cylinder is reduced, this difference of pressure causing the piston to make its stroke. As the crank shaft continues to turn the movements of the pulsator pistons are reversed, the pressures on the two ends of the drill piston also are reversed, the drill piston makes its stroke in the opposite direction, and so on continuously.

This arrangement results first of all in a great simplification of the drill and its appurtenances and a lowering of the total cost of maintenance. The usually most expensive and troublesome parts of the compressed-air drill are got rid of without any substitutes or equivalents in place of them. The drill has no valve of any kind and no valve operating devices, no tortuous air ports or minute air-valve passages, no split front head, no buffers or yielding head fastenings, no springs or side rods, not even a throttle valve. The parts thus discarded are those which are always giving out or making trouble on the compressed-air drill, and are the most expensive details to be maintained. The pulsator cylinders also, which entirely supersede the compressor plant, have no valves, either inlet or discharge, and not even a water jacket, as the cylinders have no tendency to heat up when working. The apparatus here spoken of, the drill and pulsator, it is to be remembered, comprises everything beyond the wires which connect with the generator at the power house.

The changes in operation and in the results accomplished by this novel apparatus are well worth looking into, several surprising results appearing. The electric air drill, it is claimed, strikes a harder and a livelier blow than that of the standard air drill. To this effect several things contribute. The piston is air cushioned at each end of the stroke, which the other is not, so that it starts its next stroke impelled by a certain amount of force saved and brought over from the preceding stroke. Then when it begins to move there is a simultaneous reduction of pressure in advance of the piston and an increase of pressure behind it, both conditions the reverse of those of the compressed-air drill. With the old type of drill, driven by air at constant pressure, which the drill piston runs away from, there must necessarily be some fall of pressure as the speed of the piston increases, and at the same time there must be, and for the same reason, an increase of opposing pressure in front of the piston, and a decreasing effective difference between these operating pressures. With the electric air drill the pulsator piston may be said to be chasing and gaining on the drill piston on the driving side, thus increasing the driving pressure, at least up to mid-stroke, while the other pulsator piston is running away from the other side of the drill piston and reducing the pressure in front of it.

It would seem that the critical point of the stroke of this drill, the point in the direction of the piston movement at which the actuating force culminates for the delivery of the most effective blow upon the rock, must be more precisely located in this drill than in the familiar compressed-air type, where all the runner has to look out for is to come as close as possible to the front head without hitting it too often; but if this be true the operator soon comes to know by instinct the point of best efficiency and feeds accordingly.

Perhaps the most astonishing result shown by the electric air drill is in the reduction in the amount of power required to operate it. It has been practically demonstrated that the power used, at its source in the power house generator, is only about one-third of that required by the compressor which supplies the compressed air drill of the same capacity. A 5-h.p. motor drives the electric air drill equivalent in capacity to a 3 1/4 compressed air drill, the latter when most effective requiring 20 or more horse-power at the compressor. A concise explanation is that the same air is used over and over again, its expansive force is fully employed, there are no thermal losses, there is no filling and empying of large clearance spaces.

The pulsator motor runs at constant speed, there being two different speeds available when alternating current is used and four different speeds with direct current. The drill piston makes a stroke for each rotation of the crank shaft, or at least the full air column impulse is transmitted to it. When the drill sticks in the hole the pulsator does not stop, or even hesitate, but keeps on delivering its alternate thrusts and pulls at the regular speed, say, 400 per minute, and nothing can well be imagined more effective for yanking the bit free again, and when it is released the strokes are resumed without any stopping and starting delays.

Provision is made for supplying oil to the drill as it may be required. The air carries the oil to every part, and the lubricant as well as the air is used over and over, and with a full mechanical guarantee, as we might say, that every working part will get its share.

There can be no freezing up of the electric air drill, for the double reason that it does not accumulate moisture precipitated from successive charges of saturated compressed air, and also because the working temperature of the air in the drill system never falls low enough, both high and low temperatures being avoided by the special conditions of air employment. This non-freezing feature makes the electric air drill entirely independent of climatic conditions, drills of this type being run all winter in quarry or other exposed work as well as in summer.

The drill being thus independent of climate is also quite as independent of altitude. As the air is used in a closed circuit it makes no difference to it what the external atmospheric pressure may be, and in the deepest mines or on the highest mountains accessible to man, wherever the electric wires can reach it, it runs just the same.

If there is any special line of work to which the electric air principle of drill operating is better adapted than any other it would seem to be for rock channeling. In the standard or accepted rock chamber a large air cylinder is employed, normally vertical but permitting of working to a considerable angle, and to the piston rod is suitably attached a holder which carries a gang of steels, and while the piston is reciprocating and the steels are striking the rock, the entire machine is slowly moved back and forth a determined distance, thus cutting in the rock channels of considerable length and to depths of 5 or 6 ft. or more. This work naturally requires much more power than a single drill steel and the piston in a channeler of this type being normally steam driven, the entire apparatus, a boiler and all appurtenances, is mounted upon a heavy truck traveling upon a temporary railway. Dispensing with the boiler the truck provides the ready means for carrying the pulsator apparatus, making the electric air channeler entirely self-contained and dispensing with the little carriage which otherwise is the inevitable accompaniment of the electric air drill. These electric air channelers are in use to the number of a score or more in a single quarry.

Electric Air Drills at Kensico Dam.—Perhaps the most notable job which the electric air drill has found up to the present writing is in connection with the construction of the Kensico reservoir of the Catskill aqueduct system for the water supply of New York City. The reservoir will have a shore line of over 30 miles with a storage capacity of 40,000,000,000 gal., and the dam, it is claimed will be the largest in the world, containing over 1,000,000 cu. yd. of masonry. For the construction of this dam and the related reservoir work an equipment was provided of 30 electric air drills of standard type and four of the electric air drills of much greater capacity, each mounted upon a drill wagon similâr in many respects to the electric air channelers previously spoken of.

The general direction of the dam is northwest and southeast, and about half a mile from the southeastern end a site was selected for a quarry which would yield clean, solid rock, and from this the material for the dam was to be procured. The original surface of this quarry site was covered with second growth timber which was cut off, and when the soil had been removed

Fig. 71.—Electric Air Drills at Quarry for Kensico Dam.

the glacier-scored surface was ready for the drills. No time was lost here, and Fig. 71 shows a group of electric air drills putting down the necessary holes. More than 1200 of these holes were fired at once. The material was then handled by steam shovels and dumping cars which brought it to the giant rock crushers.

The tripod drills put down holes to depths of 15 to 20 ft., while the drill wagon worked where greater depths were required to have the holes bottom near a common level.

These drills were not adopted at all as an experiment, but upon reliable evidence in advance as to what they could do, and they did it. The electric air drill complete costs more than the air-driven drill, but as the latter with its share of the compressor which drives it, and the piping and appurtenances, costs much more than the electric air drill, and involves far more first cost when installing a plant; the latter shows a great advantage. The piping and valves and other requirements of the air drill will more than cover the excess of cost of the electric air drill, and then we might say that there is the cost of the big air-compressor plant on the one side and that of the electric generator upon the other, but in so many cases, like the present, the generating plant costs nothing, because current can be furnished by the big electric companies cheaper than it could be produced by any isolated plant.

Power Cost of Electric Air Drilling.—But the most interesting and really the most important question in regard to work of this extensive character is as to the power consumption and the total cost of operating. Here we have results which can only be characterized as astonishing.

First as to the drill wagon. This drilled holes to an average depth of 30 ft. using a 5 to 5 1/2 in. starter and bottoming at 4 to 4 1/2 in., the aggregate depth of holes drilled for 8 hours ranging from 45 to 65 ft. Under test conditions 104 ft. of hole has been drilled per shift. Under the best conditions the cost with another standard type of drill would average between $0.80 and $1.00 per foot. The actual cost per foot on the electric air drill wagon is made up as follows:

Electric power for 8 hours	$0.60
Drill runner	4.00
Helper	2.50

Basing the cost per foot upon the power and labor charges alone, and considering the average daily work as 50 ft., which is very conservative, would bring the cost per foot to approximately $0.14. With a driller representing the builders of the drill this cost was cut in half.

This drill wagon is as readily handled and operated as any of the class of drills with which it directly competes, and its cost of maintenance is believed to be no greater, but the extraordinary feature is the power cost. This sometimes runs as low as $0.30 per day, and is never higher than $0.75. It is understood that the electric current in this case—3 phase, 60 cycle, 220 volt—cost the contrator $0.0125 (1 1/4 cents) per kilowatt-hour.

The power cost of the electric-air tripod drills is between 30 and 40 cents per day, drill runner $3.50 to $3.75 per day and helper $2.50. The holes are 10 to 15 ft. deep, bottoming at 1 3/4 in. diameter, and the drilling per shift is 35 to 45 ft. This would make the drilling cost of these machines based on power and labor (power 35 cents per day) approximately 20 cents per foot of hole drilled.

Comparison is suggested between the performance of these electric-air tripod drills and the standard 3 1/2-in. air operated drills. Assuming that the average free air consumption per drill per minute is 150 cu. ft., compressed to 90 or 100 lb., and taking the air from the highest type of Corliss compound condensing engine-driven compressor, with high pressure boiler, etc., would bring the power cost alone to 75 cents per day. Adding interest charges on the entire compressor installation, cost of appurtenances, pipe lines, etc., maintenance and depreciation, would bring this cost of power per drill per day to $1.25 or $1.50, or between three and four times the power cost shown for the electric air drill.

There were also working upon this contract some 3 1/2 in. air-driven drills of the type referred to above, which began work before the electric air drills were installed, and it has been found that with both drills working in the same rock the electric air drills actually averaged from 5 to 10 ft. per shift more than the air drills.

If instead of assuming the most economical type of compressor for the above comparison the air had been taken from a straight-line compressor of the old and still familiar pattern, working non-condensing, with steam below 100 lb., etc., the discrepancy in costs would be much greater. In these installations experience has made it common practice to figure the power cost per drill at $2.50 to $3.00 per day.

CHAPTER XXIII

COMPRESSED AIR FOR RAISING WATER

One of the most obvious uses of compressed air is for raising and conveying water and other liquids, and this has become one of its most extensive fields of employment. The variety of ways in which the air is applied for this purpose and the diversity of the apparatus that has been devised are astonishing. Many have nothing more than a historic interest; the actual, practical ways in which air is now employed for pumping, while differing widely from each other in efficiency and other particulars, are not numerous, yet they would be none the worse for still further elimination. The conditions under which the water is to be raised largely determine the specific device employed in the individual case, but sometimes other considerations not so defensible prevail in the selection or in the retention of deservedly obsolescent systems.

There is, according to the system adopted, much difference in the amount of air consumed as compared with the work done, but in all cases the former must be in excess of the theoretical requirement, as nothing can be done for nothing. With our present knowledge, it still pays in many cases to use air for raising water, in both small and large quantities, and as a means of permanent supply as well as in temporary or emergent cases.

It is desirable not only to do, but also to know that we are doing, the work as cheaply as possible. Our present facilities make the metering of water lifted or transferred an easy thing to do, and the raising of water, by compressed air or otherwise, can always be gaged with satisfactory accuracy, while records of such work are constantly accumulating and are accessible as guides to the engineer.

The theoretical horse-power required for raising water is:

$$\frac{\text{Pounds of water per min.} \times \text{height of lift in ft.}}{33{,}000}$$

Table XX may be taken as a starter; it furnishes the essential data as to the power required for raising water to different heights.

TABLE XX.—STATIC WATER POTENTIALS

1 Gal. per min.	2 Volume, cu. ft.	3 Weight, lb.	4 Potential h.p. in water raised 100 ft.	1 Gal.	2 Volume, cu. ft.	3 Weight, lb.	4 Potential h.p. in water raised 100 ft.
1.0	0.13368	8.355	0.025303	200	26.736	1671	5.0606
2.0	0.26736	16.710	0.050606	250	33.420	2089	6.3257
3.0	0.40104	25.065	0.075909	300	40.104	2506	7.5909
4.0	0.53472	33.420	0.101212	350	46.788	2924	8.8560
5.0	0.66840	41.775	0.126515	400	53.472	3342	10.1212
6.0	0.80208	50.130	0.151818	450	60.156	3760	11.3863
7.0	0.93576	58.485	0.177121	500	66.840	4177	12.6515
7.48	1.0	62.5	0.18928	550	73.524	4595	13.9166
8.0	1.06944	66.840	0.202424	600	80.208	5013	15.1818
9.0	1.20312	75.195	0.227727	650	86.892	5431	16.4469
10.0	1.3368	83.55	0.25303	700	93.576	5848	17.7121
20.0	2.6736	167.10	0.50606	750	100.260	6266	18.9772
25.0	3.342	208.87	0.63257	800	106.944	6684	20.2424
50.0	6.684	417.75	1.26515	850	113.628	7102	21.5075
75.0	10.026	626.62	1.89771	900	120.312	7519	22.7727
100.0	13.368	835.50	2.5303	950	126.996	7937	24.0378
150.0	20.052	1253.0	3.7954	1000	133.680	8355	25.303

It gives also the actual potential energy in the water so elevated, or the power which it should be theoretically possible for the water to develop in its descent to normal level if employed in a water-wheel or motor. When the power actually consumed in a water-raising operation is ascertained it can be compared with this table and the result will be an indication of the efficiency in the given example.

The first column gives the number of gallons of water lifted; column 2 gives the volume in cubic feet of the given number of gallons, while column 3 gives the weight in pounds of the same quantity of water. Column 4, assuming that the given quantity of water—gallons, cubic feet or pounds—is raised to a height of 100 ft. in a minute, gives the horse-power theoretically required for the lift, or the horse-power which should be developed by the descent of the water to its original level. Any other figures or quantities not in the table will be in direct proportion to those given.

The diagram Fig. 72 is based upon the same data as Table XX, but reaches farther, both as to volume of water and height of lift.

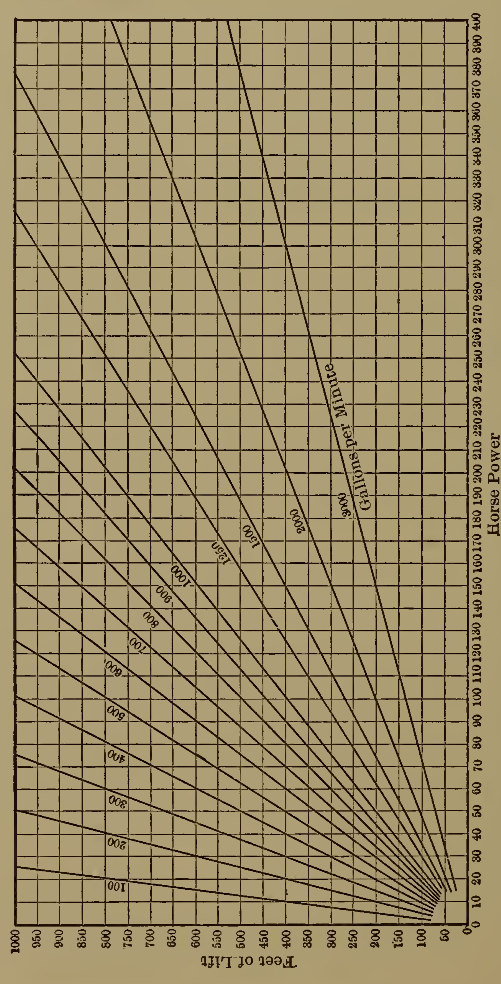

FIG. 72.—Theoretical Horse Power for Raising Water to Different Heights and in Various Quantities.

Driving Steam Pumps with Air.—First of all (as an example of "how not to do it") we may consider what it costs to drive an ordinary, direct-acting steam pump by compressed air. It is often so convenient to do this, in mines, tunnels, excavations for foundations and elsewhere, that it is done generally without counting the cost, either before or after.

The simplest case is where the steam or air and the water cylinders are of the same diameter, and have the same stroke. Here if there were no allowances to be made for clearance losses, leakages, power required to overcome friction and inertia, and if there had been no losses involved in compressing and transmitting the air—all these and other things being those which pure theory is so apt to belittle or ignore—the volume of air at the balancing pressure would just equal the volume of water delivered, and the efficiency would be 100 per cent.

The standards of efficiency which we call the possible efficiencies are really the impossible efficiencies. It is well known that they can never be attained, and very far from it in the present case. The air pressure must be enough in excess of the water pressure to overcome the frictional resistance of the machine itself, and of the water in contact with the surfaces in its restricted flow through valves and passages, and to give and maintain sufficient impulse in the otherwise inert column. For all this it will be proper to allow an initial deficiency in the air power at the pump which will average not less than 20 per cent., and the 100 per cent. efficiency with which we have started is reduced to 80 per cent.

But if there is such a deficiency, the pump will not go. This is understood; and the several deficiencies are anticipated and provided for beforehand by furnishing air in sufficient volume and at an excess of pressure to overcome them.

To run a pump comfortably the working pressure of the motor fluid should be somewhat above the actual requirement, but it is not necessary to speak of that here as affecting the power consumption.

Next, the matter of cylinder clearance may be considered, which is interesting in the direct-acting steam pump at any time, and especially so when the pump is driven by air. Nothing need be said about clearance losses in the water cylinder, for practically there are none, as all the spaces may usually be assumed to be filled solidly with water, and if everything is in good order all of the actual travel of the water piston is represented by water delivered.

Enough clearance losses in the air cylinder may be found to satisfy for both. The direct-acting pump has no crank to bring the piston to a dead stop always at the same point at the end of the stroke, and, as at the same time it must be certain that the piston shall never strike the head, very large clearance is provided. The filling of this large clearance, together with that of the unavoidable clearance spaces in the passages between the valves and the cylinder, entails another large excess of air over that theoretically required, or a deficiency of work done as compared with the air consumed of, say, another 20 per cent. This percentage of 80 being 16, 64 per cent. of the original 100 at this point is left.

There is a third great loss of efficiency when compressed air is used to drive a direct-acting steam pump, because the air is used at full pressure and shows none of the advantages of being used expansively. If, instead of using the air at its highest pressure to fill the cylinder to the very end of the pumping stroke, it could have been used in a crank-and-flywheel pump, or in one of any other design in which the air could have been cut off at the proper point of the stroke, so that it would have been discharged at a pressure nearly that of the atmosphere, then the work done for the quantity of air used would have been, on the average, varying with the initial pressure of the air, say, 50 per cent. more than without the cutoff and expansion. That is, it could and would have done one-half more work, and the failure to do this amounts to another deficiency of one-third, or, say, 33 per cent. This percentage of 64 being 21, there is left only 43 per cent. of the original 100, and this diminution of efficiency is all realized after the air has arrived at the pump, and without looking to the losses which have accumulated upon it previous to its arrival on the job.

Not to be Charged to the Air.—Now, the curious thing is that not one of the losses above mentioned, nor any portion of any of them, is in any way chargeable to compressed air. They all inhere in the apparatus and in the system by which the air is applied to the work of lifting the water.

One loss peculiar to steam and entirely absent with air is the loss by condensation, concerning which it is not necessary to present any figures. It is plain that the air should not be blamed for any of the losses of which it becomes merely the agent when driving the steam pump.

Thus far the direct-acting steam pump alone, and its deficiencies or those which it entails, have been considered. Of course, the pump has nothing to do with the friction losses in transmission from the compressor to the pump. For all such losses and for all possible leakages allow, say, 5 per cent., deducting which from the 43 per cent., found above, leaves 41 per cent., all the rest having disappeared after the air left the compressor. The transmission loss just allowed would be greater with steam, while some of the distances which air may be carried are prohibitive with steam.

While the cost of compressing the air used is not considered, attention might be called to the approximate figures for compressing free air to, say, 80 lb., gage. Assuming a steam-driven, reciprocating air compressor and both compressor and pump to run regularly under their respective rated loads, the power which is being developed in the steam cylinder may be taken as a starting point or the basis of efficiencies. First must be deducted a sufficient allowance for the friction of the entire machine, and for the leakage and clearance and other air-cylinder losses, amounting altogether to an inefficiency of at least 20 per cent., leaving 80 per cent. Then in the compression of the air the excess of power required for the actual adiabatic compression, instead of the theoretical isothermal compression, will be 34 per cent. This percentage of 80 being 27, the surviving efficiency will be 53 per cent. This being the efficiency of the compressor and 41 per cent. the pump efficiency, the ultimate efficiency of the combination is 21.73 per cent.; that is, it will take about 5 h.p. in the steam cylinder of the compressor to realize 1 h.p. in the actual lifting and delivery of the water. Actual results are seldom any better, and often much worse, than this.

At least one hint may be taken from these figures, which is that the loss is the least where the working air pressure is the lowest. In the case of the direct-acting pump the difference may be made in the relative capacities of the air (driving) and the water (driven) cylinders, and, in general, the larger the former is, the better. Thus supposing the air to be used at 40 lb., gage, instead of 80, the excess of mean effective resistance in the air compressing cylinder in the act of adiabatic compression would be only 22 per cent. instead of 34, and the loss in the air cylinder of the pump through not using the air expansively, would be relatively the same, so that there would be a saving at both ends.

Direct-displacement Pumps.—Naturally, the first arrangement thought of when it is proposed to use compressed air for raising water is that in which the water is displaced volume for volume by the air, this air being compressed to a pressure corresponding to the height of the lift, and this scheme has been worked up in a great variety of ways, but with little variation of result.

Fig. 73 shows diagrammatically a pumping apparatus of this type. The submerged chamber here shown is assumed to have been filled with water and this water is now being expelled by air pressure. The air comes direct from the compressor, passes the three-way valve and enters the top of the chamber, its pressure driving the water up the vertical pipe at the right. When the water is all expelled, the three-way valve is changed to the other position, shutting off the compressed air and opening communication between the interior of the chamber and the atmosphere and allowing the air to escape. Then the valve in the water pipe at the upper right-hand corner of the submerged chamber closes by its own weight and the pressure of the water above it, and the water rushes into the chamber through the valve at the bottom. A reversal of the three-way valve to the position shown puts the compressed air at work again and the content of the chamber is expelled as before, and so on. The three-way valve is usually actuated by a float, making the operation automatic. There are generally two of these displacement chambers which work in connection, the same valve-operating mechanism serving for both, and thus as one is filling while the other is discharging, the water flow is almost continuous.

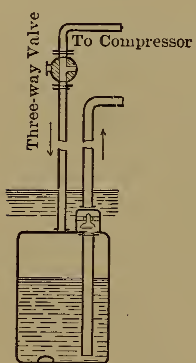

Fig. 73.—Direct Displacement.

It is well understood that the displacement pump does not use the air to good advantage, but for economy it is still much better than the air-driven, direct-acting pump. There is no friction of reciprocating parts and the clearance losses are very small, so that instead of the 20 and 20 per cent., which is allowed for the direct-acting pump, a single 10 per cent. will fully cover these items, leaving at this stage 90 per cent. instead of 64. The loss for the same pressures in not using the air expansively would, of course, be unchanged, and therefore the 34 per cent. which was assumed would still apply, and 34 per cent. of 90 being 30 there is 60 per cent. efficiency for the displacement pump instead of the 43 per cent. of the air-driven, direct-acting pump, and 57 per cent. instead of 41 per cent. when the air leaves the compressor.

The economy of the displacement pump is much better where the lift is small, which makes it specially applicable for transferring sewage and similar service, especially as the water may be loaded with a large proportion of solids and semi-solids without impairing its action.

Return-air Pumping.—The progress of invention in any line seems to be one of gradual but continuous revelation and development. Every device, so that it will work at all, seems to have a right to be completely tried out, and to have all its possibilities and its defects revealed before it can expect to be and actually is displaced by something better.

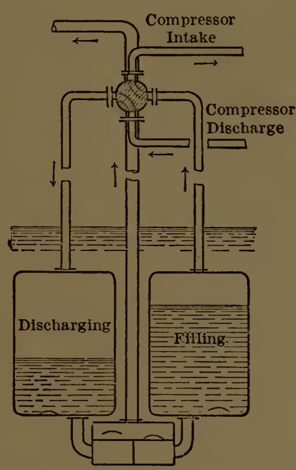

FIG. 74.—Return Air Pumping.

When the power-wasting and otherwise objectionable direct air-pressure, water-raising apparatus has had its every chance and has been found wanting, then, and not until then, comes another device which is at once hailed as just what is needed. The principle of it is at once self-evident, and its perfect success when applied requires no practical demonstration.

This is the return-air system of compressed-air pumping. Its operation, in contrast to that of the direct-displacement pump is that while the old pump will take all the air that is given to it for raising and delivering a given weight of water, and will return absolutely nothing of what it receives, this new device will take only the pressure and volume of air which exactly pays for the work done, and, so to speak, it returns all the change.

The whole thing can be simply explained, the diagrammatic sketch in Fig. 74 perhaps assisting. It must be premised that a return-air system is not a cheap and handy little device ready to go, for instance, wherever a steam pump might be employed, and to do the same work; it is intended for large and permanent installations, each plant being complete in itself and doing no other work.

The Apparatus Used.—There is, first of all, an air compressor whose entire business it is to operate the one pump, the capacities of the pump and compressor being adapted to each other. There is nothing special about the compressor except that it is a single-stage, reciprocating machine, and it, of course, may be driven by any available style of power transmission and application.

The pumping part of the apparatus consists of two similar chambers to be alternately filled with water and emptied by the expulsion of the water to the level required. These chambers are submerged in the water to be pumped, or at least are located near and below the lowest working level of it, so that the water will always be able to flow into either chamber by gravity. The compressed air is conveyed to the pump from the compressor by separate pipes to each water chamber, and each pipe also leads the air, after its work of water expulsion is done, back to the air intake of the compressor. It is thus a closed system, and the same air is used over and over again, provision being made for continually replacing the little that may be lost by leakage or by absorption in the water. The valve arrangements are clearly shown in Fig. 74.

Operation.—Assume that the pump is actually at work, that one of the chambers is full of water, with sufficient air under pressure flowing in at the top of the chamber, exerting its force upon the water and driving it through a retaining valve and up a water pipe to the point of delivery.

Thus far the operation is precisely the same as with any displacement pump, and when the water is all, or nearly all, driven out the chamber is full of air at the pressure required and determined by the height of the water delivery. The regular operation by the old way would now be to close the communication with the compressor and to open another to the atmosphere, allowing the air, expanding as it goes, to escape without doing any more work and leaving the chamber filled with air at normal atmospheric pressure. Water is then allowed to flow in and to fill the chamber again, which drives out the remainder of the former charge of air, and this chamberful of water is driven out and up by another charge of full-pressure air as before, and so on.

With the return-air system, however, the air is not let off so easily, but is retained to do a lot more work. The air is discharged from the water chamber and up the air pipe which, by the action of an automatic or a positively operated switch, is now in connection with the intake instead of the discharge of the compressor, and whatever pressure this air may have is thus always being exerted upon the intake side of the compressing piston, and the pressure upon this side overcomes an equal amount of resistance upon the compressing side of the piston, or it deducts just so much from the total power required to drive the piston and to do the work of re-compression.

Using the air in this way, its entire expansive force is retained upon the credit side of the system. Expansion continues until the air-pressure is lower than that of the water which is waiting to enter, the difference of these pressures depending upon the speed at which the system is being driven, and then the water enters and drives the remainder of the air into the compressor intake at this pressure, instead of letting it further expand, and thus raises the total mean effective assisting pressure at the back of the piston above that which would have resulted from the expansion alone.

Each return-pipe installation is an individuality, with special conditions to be satisfied and calling for special sizes and capacities, both relative and actual. The system is advantageously applicable to both extremes of the possible range of practical requirements, and to all between.

Constant Lift System.—Where the lift is practically constant, as in raising water from a lake or river to a water-works reservoir, the compressor and the pump may be so adapted in capacity as to utilize all the expansive force of the air with comparatively little of the follow-up pressure from the water, but still with the highest possible ultimate efficiency and economy. In any case, as the compressor is in immediate touch with its work there is no loss from overcompression or from compressing above the actual requirement, as is often the case with the direct-displacement systems.

Variable Lift System.—If the return-air system is employed where the level of the water intake has a considerable variation of height, as in emptying a large, deep shaft of a mine, the follow-up pressure of the water in refilling the operating chambers will vary constantly with the water-level, and at all levels will approximately adjust the work of the compressor according to the actual water lift at the time; and, although the water must always be lifted under its full actual head at the time, and with the requisite overbalancing pressure for the expelling air, the compressing of the air to that pressure will require little or much power according to whether the return, or intake, pressure at the compressor is much or little, corresponding to the height of the water-supply. With the direct displacement, and no return of the air, there would be none of this compensation, either from the re-expansion of the air or from the follow-up pressure of the incoming water, and all the water lifted, whether the level of the supply was high or low, would require the maximum and entirely uncompensated pressure to raise it.

An Example of Application.—There was a case where it was necessary to provide, and to have constantly ready the means of emptying a shaft of considerable area, more than 300 ft. deep and normally full of water, with a tunnel at the bottom of the shaft also to be emptied. This will be recognized at once as a problem

Fig. 75.—Return Air Installation at Harlem River Siphon.

easy enough of solution, in a way, but not so simple when speed and economy were to be considered. There had been actually installed for this work a hoisting engine and a large cylindrical bucket, say 4 ft. in diameter and 12 ft. long, to be successively filled, hoisted and emptied.

This might strike one as a rather primitive arrangement, but it had at least the merit of using the power with reasonable economy. The varying height did not at all affect the weight of the bucket of water, and the power consumed was therefore always directly as the height, and with engines showing good steam economy the arrangement was not wasteful of power.

This bucket arrangement was actually used for emptying the shaft and tunnel once and it worked all right, but nevertheless it was superseded by the return-air system, which could show good power economy and could do the required work in much less time, which in this case was the all-important consideration.

The shaft here spoken of is known as shaft 25 of the "new" Croton (not the Catskill) aqueduct, and the tunnel is the portion of the aqueduct which crosses under the Harlem River, 300 ft. below its surface, near Washington Bridge, New York City. Fig. 75 shows the return-air pump house over shaft 25.

Valve Arrangement and Operation.—In the foregoing outline of the operation of the return-air system reference has been made only to the action in one submerged operating chamber. It is understood that there are two similar chambers operated alternately, one filling while the other is emptying, so that the lifting is practically constant. The four-way valve or switch, as it is called, for reversing the connections of the two pipes at the proper times is operated in different ways according to the work. For a straight, steady water-works job the switch may be operated mechanically, tripping one way or the other for each given number of revolutions of the compressor; it may be operated automatically by the fluctuations in the level of the water supply, and it may also always be operated by hand. It is not necessary to completely fill either chamber before reversing, because the unused pressure is not lost in any case, and there is nothing corresponding to the clearance losses in the direct-pressure ejection system.

The actual installations of the return-air pumping system already show a wide and interesting range of successful employment; what the device needs most of all is to be better known. It seems to decline no job which is possible by any system, it works for the lowest air output, and is especially successful in many lines which are not permissible to the formal mechanical devices. It pumps, for instance, water which is so charged with solid matter that it can only be called semi-liquid. It is in some cases regularly employed for conveying the solids so contained, as sand or marl, the mixture when so raised going to settling tanks where the water is allowed to flow off, leaving the solid portion to be shoveled or otherwise handled and conveyed wherever required.

CHAPTER XXIV

THE AIR LIFT

One of the most important, and now also among the most extensive and most rapidly extending employments of compressed air is in the so-called "air lift," used for the raising of water and other liquids. This name is not correctly suggestive of the characteristics of the device, but it seems to be quite firmly fixed and no one seems to be able to supplant it with anything better.

The water-works of many towns and cities in the United States and elsewhere are entirely dependent upon the air lift, and its use for the obtaining of water from its natural subterranean sources is rapidly extending throughout the civilized world. It is used largely for the pumping of oil wells, for the unwatering of mines, and it handles semi-liquids, or liquids carrying a heavy burden of comminuted solid matter, with the same facility as clear water, making the water thus in many cases the cheapest conveyor available.

The adoption of the air lift for any specific service is seldom determined upon by considerations of power economy alone. It has its special adaptations to some exacting conditions, and it may be said to hold its own and to be continually extending its holdings by the fact that it can do work to which no other devices can be applied. The wide range of its employments, and its continuance in them is the best possible evidence that after all it pays. It is also free for anyone to employ it, as there can be no monopoly of the principle involved.

It is not necessary to make any mystery of the operation of the air lift, and no one has any privilege to pose as an exclusive repository of special information concerning it, although in the practical applications of it experience counts for more, and is more absolutely necessary than in almost any other field.

Principle of Operation.—The essential principle involved seems to be very simple and to be easy of explanation. Say that we have an open well, or in fact any body of water, so that it be deep enough, and regardless of its volume or what may enclose it, a

Fig. 76.—Artificial Waterfall from an Air Lift.

pond or lake or river being as good for our purpose as a well of whatever dimensions; and say that in this water is suspended or fixed a vertical pipe with open ends, the lower end extending a considerable distance below the surface of the water, the water being thus perfectly free to enter the pipe. Under the conditions, the water will rise to the same height inside the pipe as it is outside, and when the water-levels coincide there will be nothing operative to cause any movement of the water either upward or downward.

If we wish to induce the water inside the pipe to rise higher than the water outside the pipe it can be done by reducing the weight, or more correctly, the specific gravity of the column of water as a whole contained within the pipe, and then the greater relative weight of the water outside the pipe will act to force the lighter column of water upward. The simplest way in the world to lighten the column of water in the pipe would seem to be to mix air with it, and this the so-called "air lift" does, and it is practically all that it does, while gravity does the rest.

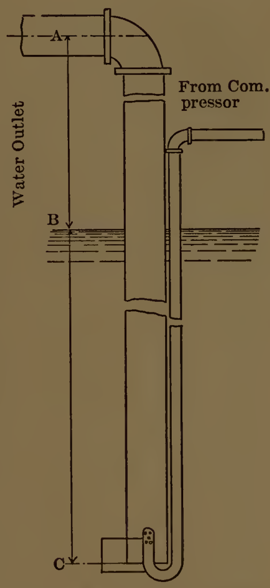

FIG. 77.—The Principle of the Air Lift.

Say that, with our pipe standing in the water as above assumed, and with the water standing at the same height within it as outside, a compressed-air pipe considerably smaller than the other is run down alongside with its lower end turned up and reaching a little distance up into the large pipe, as in Fig. 77. In the placing of this air-pipe it may be supposed that the water has entered it and has risen in it to the normal water-level, if there has been no confined or compressed air in the pipe to oppose it. If now compressed air is turned on to the small pipe with a pressure sufficient to drive the water down to the lowest point in the bend of the pipe, the pressure required being determined by the height of the water surrounding the pipe and this pressure, 1 lb. for each 2 ft. (speaking loosely) of water depth, being the greatest pressure that will be called for in the operating of this particular lift, then as soon as the bend in the pipe is passed the air will escape upward out of the air-pipe and will diffuse itself through the column of water above.

We must not go too fast here. The air "lift" has not yet begun its lifting. The diffusion of the air through the column of water elongates the column of water upward, but does not yet lighten it as a whole. The air might go on mixing with and diluting the water and gradually working its way up through it and finally, if the pipe was long enough, the air would escape at the top of the column as fast as it was admitted at the bottom, but still there would be no "lifting" of the water by this operation alone. This gradual working of its way up through the water which the air thus does in the stationary column goes on also, but not so rapidly, when the column is moving and when the lift is in full operation, this constituting the "slippage" which seems to be one of the unavoidable sources of loss in the air lift.

As a matter of fact in the air lift the air does not work its way up through a column of water of unlimited length and escape out of the water at the top, leaving the water inert. Before this height is reached the water pipe ends and there is a continuous discharge of its contents. This discharge of the upper portion of mingled water and air at once reduces the weight of the column and then it begins to be pushed upward by the heavier solid water which is now able to force its way in at the lower end. The escape or discharge at the top makes the column continually lighter than is required to balance and entirely resist the in-pressing water below, so that the solid water continues to flow in, and then the entire column is kept rising and flowing off at the top, as long as the supply of compressed air is sufficient to maintain this condition of unbalanced pressure; and there we have the air lift complete. It is ultimately worked as we see entirely by the force of gravity.

It will be better to finish with the air and its career in the water before considering the action of the resulting unbalanced pressures. The levitation function of the air begin understood, it will be evident that it must be desirable to have it mixed as intimately as possible with all the water in the column, so that here ingenuity and invention may play an important part in providing means for scattering the air and mixing it at the begining all through all the water. If such scattering and intimate mixing of the air with the water does not occur at the instant of its discharge from the air-pipe no such result can be possible later. It is to be assumed that the air will be distributed through the water in bubbles, large or small, and it is better that they should be as small and as numerous and as nearly as possible at uniform distances apart rather than to have them fewer and larger and irregularly distributed.

The originally propounded theory of layers of air alternating with pistons of water in the pipe represents an absurd impossibility. Whatever the number and size of the air bubbles at the beginning they will always be merging together to form fewer and larger ones. Whether the aerated column be stationary, as first assumed, or moving upward with the lift in full operation, the levity of the column which the air is employed to produce will be determined by the amount of air actually distributed through it during the period of its upward passage through the pipe. If some of the air escapes by working up through the column, instead of moving with it, it represents power lost, and this loss will be greater with large bubbles than with small, because they will always float up through the water at the greater speed.

Air Used Isothermally.—In its operation the air lift uses the air expansively and gets the benefit of the expansion. When the lift is in full operation and the air and water mixture is flowing at a uniform speed through the discharge pipe there will be at any instant a given quantity of water and the equivalent of a given volume of free air. The air actually will not be either free air nor air at the full working pressure. When it is first discharged from the air pipe and mingles with the water it has already lost a small portion of its pressure, and as it rises with the column it will expand as the pressure due to the depth decreases, and it will be discharged with the water at a pressure little above that of free air. As the value of the air in the column is in the quantity of water it displaces, this displacement will be greater per volume of free air at the top of the column than it will be anywhere below it. As the air is so intimately mixed with the water it must be assumed to take the temperature of the water, or its expansion may be said to be practically isothermal, which is as profitable to the user of the air when expanding as the absence of this condition is unprofitable when the air is being compressed.

Compressed air in the air lift, it has been seen, does not push or force the column of water upward after the lightening of the column, the latter being its only function, and the flow upward is caused by the unbalanced weight of the solid water outside and below the pipe. This suggests at once the necessity of maintaining a sufficient preponderance of pressure from the inrushing water, and this is only secured by the submergence of the pipe or its extension to a sufficient depth below the surface of the water. The height of the lift is the principal factor in determining the submergence required, the invariable condition being that the higher the lift the greater shall be the actual submergence. The percentage of submergence, spoken of later, is another matter.

In practice the matter of submergence assumes the greatest importance, and mistakes or misjudgments in this particular are not infrequent. In planning an air lift for a bored and tubed well wherein the water does not rise to the surface but stands constantly at a certain distance below the surface, it is comparatively easy to compute approximately the submergence which should be adopted for delivering the water at a given height, experience rather than theory dictating the figures; but if when the pumping begins the height of the water in the well falls any considerable distance, as it often does, then the proportionate and the actual submergence may be quite inadequate. The submergence will be reduced as many feet as the surface of the water supply falls, and the lift will be increased an equal distance, a double-acting cause of incompetence. In many cases where a new air lift is installed provision is made for lengthening or shortening the pipes to get the best submergence according to the developed working conditions which could not have been anticipated.

With a given submergence the lift may be greater or less, within wide limits, and the lift will still work, but there is for each case an approximate ratio of submergence to lift which gives the best results, both as to the volume of liquid raised and also as to the air cost, or power cost, of raising it. With a fixed submergence, whether the lift be proportionately small or whether it be as high as possible, the air-pressure required will be the same, but while the pressure is thus determined by the submergence the actual volume of air per unit of time, and also the volume of water raised will be influenced and determined by other conditions, especially by the actual volume of air supplied and the resultant rate of flow of the water.

Details of the Air Lift.—In computations upon the air lift the submergences are stated in percentages. The submergence percentage is that portion of the whole length of the discharge pipe which is submerged, the remaining percentage constituting the lift. Thus with a lift of 100 ft., if the submergence is 60 per cent., the lift, or 100 ft., will be 40 per cent. The 60 per cent., the submergence, will therefore be 150 ft., and the total vertical length of the water or discharge pipe will be 100+150=250 ft.

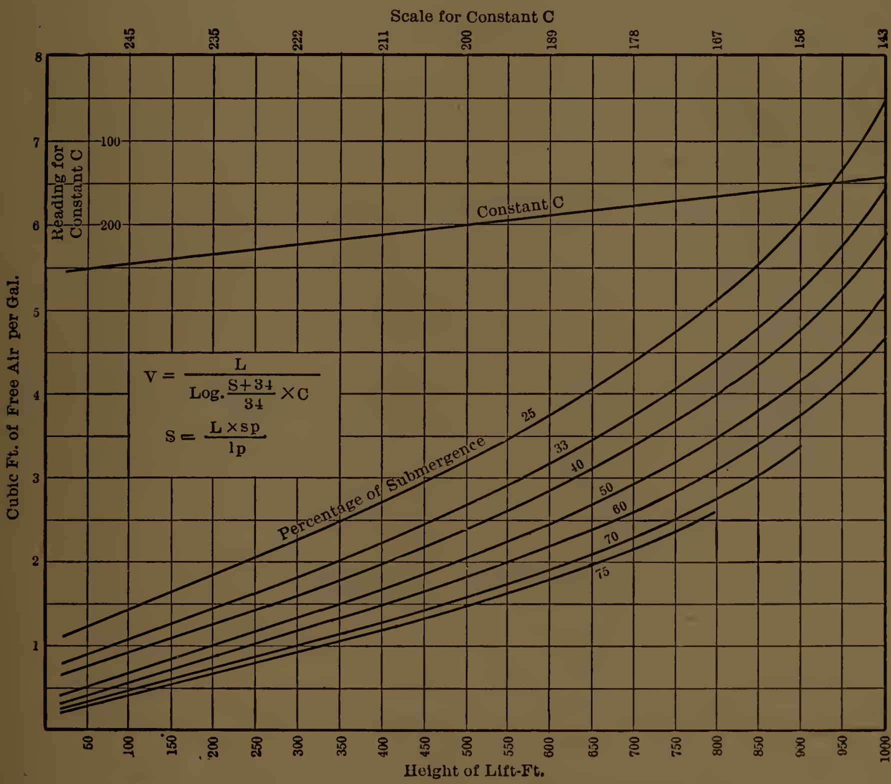

Fig. 78.—Free Air Requirements for Different Submergences.

In Fig. 77 AB is the lift and BC is the submergence, both being stated in percentages of the total length AC.

Attention is now directed to the chart or diagram, Fig. 78 from which may be readily determined the volume of free air required with different percentages of submergence to raise 1 gal. of water to different heights from 20 ft. to 1000 ft. The data embodied in this diagram are practically the same, except in one particular, as the figures of a table given to the public a few years ago by Mr. E. A. Rix, of San Francisco, who has done much in various ways for general compressed air practice, the table having been compiled by Mr. Geo. H. Reichard of the same city.

The formula by which the table spoken of was computed is as follows:

$$V=\frac{L}{\text{Log.}\frac{S+34}{34}\times C}$$

Here V is the volume of free air in cubic feet, this being taken in this case as the actual piston displacement of the compressor.

L is the lift in feet.

C is a constant, 234, unchanged throughout the computing of the table.

S is the actual submergence in feet, which is obtained as follows:

$$S=\frac{L\times sp}{lp}$$

sp being the submergence percentage and lp the lift percentage.

Say that the lift is 100 ft. and the submergence 60 per cent., then:

$$S=\frac{100\times 60}{100-60}=150$$

Using figures now in the original formula we have:

$$V=\frac{100}{\text{Log.}\ \frac{150+34}{34}\times 234}=0.584\ \text{cu. ft.}$$

For the air-pressure required for any lift, and with any percentage of submergence, it is convenient to divide the actual submergence in feet by 2 to get the gage pressure in pounds. This gives enough pressure in excess of that due to the water head to allow for the pipe-friction and other losses.

The one particular in which the computations for our diagram differed from those for the Rix table spoken of was in the constant C. This, as indicated by the line in the upper part of the diagram, instead of remaining the same, 234, all through, was made to decrease gradually as the lift increased. Thus for a lift of 200 ft., our constant is 235, practically the same as that used by Mr. Reichard, but for 100-ft. lift it is 245, for 500-ft. it is 200, and so on. The result of this change is to increase more rapidly the free air per gallon as the lift increases, which seems to agree better with observed results.

The line for constant C in the diagram follows very closely the figures given by Mr. H. T. Abrams in a lecture at Columbia University.

The most striking thing observable in an examination of this diagram is that the volume of free air required per gallon of water lift constantly decreases as the submergence increases. Thus at 25 per cent. submergence and 200-ft. lift, the free air required per gallon is 1.83 cu. ft., while for the same lift and 60 per cent. submergence the free air is 0.88 cu. ft., or less than one-half. In the first case, however, the air pressure required is about 32 lb. gage, while in the other case the pressure would be 140 lb., and the mean effective pressure required for the compression of the air (adiabatic) to 32 lb., would be about 20 lb., while for compressing to 140 lb. the mean effective would be 50 lb. Then comparing $1.83 \times 20 = 36.6$ with $0.88 \times 50 = 44$, it appears at once that the apparent cheapness of deep submergence is nullified or reversed, and other considerations determine what submergence is best.

While, as was said farther back, the higher the lift the greater must be the actual submergence, the percentage of submergence will be the other way. It has been recommended in some quarters that a submergence of 60 per cent. be adopted in all cases, but better results are obtained by departing from this in both directions, going to 65 per cent. for lifts of 20 ft. or so and down to 40 per cent. for lifts of 500 ft. or more to secure the greatest delivery. This is in recognition of the augmented pipe friction when the vertical length of pipe to be traversed is so great.

The air lift is intended for steady work and not for occasional and frequently intermitting service. It is not to be stopped and started at any odd time like a common reciprocating lifting pump. Its efficiency is largely influenced by the correct adaptation, or otherwise, of the sizes of air-pipe and discharge pipe to each other and to the rate of air delivery, the latter being entirely controllable, while the inflow of the water adjusts itself to the conditions provided. Nothing is thought of here but the relations of discharge pipe and air-pipe to each other, it being assumed temporarily that the water is free to flow into the bottom of the discharge pipe as the "air lift" takes it away. The speed of this voluntary (as we may call it) and uncoercible flow of the water really determines the best working speed, and also, within certain limits, the possible working speed of the air lift. As might naturally be supposed, the rates of flow in the pipes and the corresponding actual deliveries of water will be more rapid for the low lifts and small actual submergence than for the high lifts and deep submergence, the greater length of pipe causing the greater retarding friction.

While from what has been written above it would seem that almost any one might install and operate an air lift, there is probably no application of compressed air in which experience counts for so much, and it is recommended that in all cases a competent expert be consulted or employed. In this view of the case it odes not seem worth while to lengthen this article by the insertion of tables of pipe capacities and other working data, as these may be obtained from builders of air-compressors and others in touch with the business.

The writer is tempted, however, at this point to offer a suggestion as to a novel application of the air lift which seems to be altogether feasible and which it may be worth while to think and talk about.

FOR A CATSKILL AQUEDUCT MONUMENT

Probably few persons have any adequate and abiding appreciation of the fact that the great engineering works of the day are so largely indebted to compressed air for their existence. This is especially true of the great aqueduct which is to bring the water of the Catskills and distribute it over the entire area of Greater New York. It is not now possible to see how without the sustained activity of the air-driven rock drill the aqueduct could ever have been.

That this is the simple fact is indisputable, and it would seem to furnish us sufficient warrant to speak of the great aqueduct in Compressed Air Practice, especially as we here purpose to lay it under additional obligation to pneumatic agencies.

It is seriously proposed, and the project seems to be under way, to erect an imposing monument to the Panama Canal. The gratuitous inadequacy of the proposition resides in the fact that the great work is in itself all monument. It is all open to the view, and both in detail and in its entirety it tells its own story, impressing the beholder with its magnitude in a way to belittle and render futile any artistic or architectural commemoration that could be devised.

The same will be true of the New York State Barge Canal when completed, of the tunnels and subways of New York City, and of the city itself, when it is finished. Rome required no monument besides itself to perpetuate its renown.

There are, however, other great and costly works for the service of the people which are in no respect spectacular, or even visible at all, and which have no opportunity to appeal to the eye with any impressive reminder. This is especially true of our modern pressure water tunnels which give to the world no sign of their existence, and which it is easier to forget than to appreciatively remember. The aqueducts of the ancients were eminently monumental, and when they survive they suggest adequate conceptions both of the works themselves and of the communities which built them.

The original Croton aqueduct, which followed the ancient practice in providing a level flow for the water, has a worthy and enduring monument for itself in its own "High Bridge" which carries it across the Harlem River. The "New" Croton aqueduct, which is a much greater work, but which, being a pressure tunnel, is all underground, has nothing to show but a gate house.

The Catskill aqueduct, of whose magnitude and magnificence as an engineering work it is idle to speak, also functionally provides no monument for itself; but what material achievement of man could more loudly call for a worthy visible and enduring reminder to compel the appreciation of its myriads of beneficiaries who may be mostly yet unborn?

The only appropriate type of monument compellingly suggests itself at once. Nothing can be thought of but a fountain.

It happens that a fountain regardless of its specific commemorative function in each case, is the most effective and satisfactory of all monumental devices. No better work has been done by the artist-architects of the world than in the designing and erecting of fountains, and these works are the most conspicuous and the best remembered of the sights which are the boast of the great cities of the world.

Too often it has been that in the fountains which should have been most admired and renowned there has been too much art and too little water. Magnificent erections of stone have been provided with imposing statuary in marble or in bronze, and one or more little trickling streams of water, and many times the sight-seeing traveler has to turn away in disgust when he finds these fountains dry.

The cost of the erection of the fountain ends with the completion of the work, but that of the water to animate it continues. Water costs for its storage and conveyance, and frequently also there are constant pumping charges, and after the water has performed its function in the fountain it all runs to waste, so that the cost of the fountain may be said to begin rather than to end with the first flow of the water.

Now, in the general water supply scheme of Greater New York, as backed by the Catskill aqueduct, there is a unique opportunity as regards the providing and maintaining of a spectacular commemorative fountain. Here may be a fountain which will be a novelty to begin with, in that it will be chiefly if not entirely a fountain of water, and not a fountain of stone or of bronze with a little water to save the name of it. The water for the fountain may be in practically unlimited quantity, it will cost nothing for its flow at the fountain, and none of it, except the little that evaporates, will be diverted from the ultimate channels of consumption. All the art called for from the designers of such a fountain would be for the spectacular arrangement of the water flow, but in that might be found an opportunity such as never was before.

All this is so simply because the flow of the aqueduct is to come into the heart of the city with a pressure much above that which can be used for the lower or general service. When the water enters the distributing reservoir—say the present large reservoir in Central Park—let it, all of it or as much of it as may be required, enter by an overflow at such a height as the full pressure will give it, instead of by a subterranean and restricted channel; and there we have our fountain. Not a word will be added here as to the details of it, if only an unassisted gravity fountain is to be considered, and even that should make a very satisfying exhibit.

But so much better than this could be done. The possibilities of the air lift for fountain purposes, it is believed, have never been exploited. The water discharged by the air lift is not, as we might say, "solid" water, such as would be delivered by a mechanically operated pump, but is water mixed with much more than its own volume of air, and consequently the discharge for any given quantity of water per unit of time with the air left is at a much higher velocity than it would be with the water unaerated.

Fig. 79 is a noble stream of fully aerated water from a 6-in. air-lift pipe. The same quantity of water coming from a steam pump of the same capacity would not show nearly the life or the bulk of this stream. The pump stream would show the color of the water, and if not too oily it would be more or less transparent

Fig. 79.—A Typical Air Lift Discharge.

This stream, as we see, is beautifully white with no suggestion of transparency. Not only is the visible volume and velocity increased by the air lift, but the jet has a light and feathery effect which cannot be produced by any other means. Fig. 80 gives some idea of the fairy lightness of the air-lift stream, although the photo was not taken for the purpose of showing this, and better effects may be produced where a suitable background is provided for the display. This suggests the necessity of properly locating the fountain we are suggesting and the position for it might not be in the middle of a large body of water, which is the situation which has frequently been chosen.

Fig. 80.—Fountain Effect of Air Lift.

But it is idle to speak of the display which might be made with air-lift jets, or with such jets in connection with gravity discharges, unless we have a sufficient supply of compressed air for the lifts. This also the water of the aqueduct could easily be made to provide in its necessary transmissions from the higher to lower pressures, with no cost except for the construction of the compressing plant; and this might be located wherever most convenient. As appeared in our chapter on the Taylor air-compressor, any fall of water, whatever its height, can be made to compress air to any pressure desired, the only imperative condition being that the volume of falling water shall be sufficient for the work. The Catskill aqueduct seems to be well provided with both the water and the power for a monumental fountain worthy of itself and of the Metropolis.

CHAPTER XXV

AIR FOR LARGE STEAM HAMMERS

The use of compressed air for driving large hammers such as are normally steam driven, in boiler making, ship building, bridge work, and general heavy forging is rapidly extending. Not only are many original steam hammers now so operated, but they are always operated with satisfaction if not always with computable and demonstrable power economy, and when once air is employed for the purpose its use is rarely if ever abandoned. The considerations urging this use of air are quite convincing.

It has come in my way to know something of the manufacture of rock drills. These drills are built to be operated some by compressed air and some by steam. A steam drill, by the way, may always be operated by steam, but an air drill cannot always be operated by steam on account of its destructive effect upon leather packing. Every drill is tested or, as we might say, broken in, at the factory before it is sent out. This testing operation is much more thorough and exacting than the term would suggest. Each drill is run for a considerable time under different conditions of stroke, pressure, speed, etc., until it is found to work correctly under all normal conditions.

It happens that all steam drills while thus being tested are run first of all with air and after that with steam, and after the drill has been made to work all right with air it still requires time and nore or less coaxing before it works equally well with steam, the latter being always the more difficult proposition. The rock drill is only a smaller steam hammer, and the testing experience with the drill is closely typical of general experience with the steam hammer.

It will readily be conceded that the use of steam in the steam hammer is never without a number of objectionable accompaniments. I speak now of the hammer as installed under not unfavorable conditions; it may be located not far from the boiler and freely supplied with comparatively dry, live steam. The piston or tup is a solid mass of metal, its weight being largely depended on to give force to the blow, and the cylinder also is much heavier than that of a stationary engine of the same diameter, and when the steam is turned on its first work is to heat up all this mass of metal, thus involving not only the condensation of a quantity of steam, but also the flooding of everything with water. No matter how perfect may be the arrangements for taking care of the water, it still works out of the stuffing boxes, drops around when it is not wanted, and is the familiar and constant nuisance of the steam operated hammer. The hammer also is never operated continuously, so that this warming-up and steam condensing operation is repeated more or less every time the hammer is started up. The cost of steam wasted by condensation is quite an appreciable addition to that of the steam employed for the working. There are expansion troubles also connected with the use of steam, the parts not heating up and expanding equally so that the warming-up process every time the hammer is operated also requires the playing of the hammer up and down, the working of the valves to have them free, etc., and besides the steam consumed considerable time is required. This does not necessarily imply any delay of the work, as the hammer may be got ready ready beforehand, but it takes the time of a man who might be doing something else.

Steam thus charges continually for waiting in readiness as well as for the actual work it does, while compressed air costs nothing except for work actually done, and this it is always and instantly ready for. The hammer the first thing in the morning is readier to go with full force the instant the air is turned on than it is with steam after fifteen minutes of warming and limbering up, and the same warming up is required more or less every time the hammer is operated.

Another important point is the lubrication of the hammer. With air the oil remains on all the working surfaces the same as with machinery which is all exposed, while with steam the oil disappears almost immediately and lubrication must be almost continuous and requires constant watching. This reliability and constancy of lubrication in the one case and the precarious uncertainty of it in the other is especially brought out in drill testing.

Three or four different samples of lubricating oil were once sent to the drill testing department above spoken of for trial in the drills, with the purpose of ascertaining the special adaptability of the different oils for use with steam or with air respectively. Very little satisfaction resulted from the trial. Whichever sample was tried upon either drill, the uniform result was that when the steam drill was taken apart and examined all the surfaces assumed to be lubricated were "as dry as a bone," the oil having entirely disappeared, while the similar parts of the air-operated drill were all quite oily. Precisely the same results are observed in the lubrication of large hammers accordingly as they are driven by steam or by air.

In considering the matter of air for steam hammers it will appear all the way through that it is not to be settled by merely comparing the costs at the boiler or the power house. The constant readiness, the handiness and liveliness of operation, the saving of the time of the workers at the hammer, outweigh in each individual case many pounds of coal.

When the hammer is operated at a distance from the source of power, the advantage in the use of the air is more pronounced. A steam pipe is of course losing heat and condensing steam all the way along, and the steam is wet and heavy when it gets to the hammer, while there is practically no loss in the transmission of the air, and absolutely no difference in its working readiness at the hammer.

There was recently a specific case up for consideration in which if steam was used it would be necessary to pipe it 1800 ft. The loss by steam transmission even with costly heat insulation would be quite large for this distance. In the case of the air there might be a fall of pressure of 1 or 2 lb. this loss being easily computable when the conditions are specified, and the loss of pressure would be almost entirely compensated for by the corresponding increase of volume delivered. The loss by leakage, assuming the piping to be properly laid, would be so small as to be entirely negligible.

We give here what data we have immediately available as to the air required for operating a steam hammer. We have information of many plants where steam hammers are driven by compressed air, but in every case some of the air is used for other purposes, so that it is not possible to get the actual air consumption in any given case, and this would be difficult of ascertainment for purposes of comparison in any case, on account of the intermittent use of the hammers and the difference in the total time of employment in each case as compared with any other.

As a starting item we may note the statement which I have from responsible and experienced hammer builders that the largest amount of free air required for the continuous running of a steam hammer is 26 cu. ft. per minute compressed to 90 lb. for each nominal 100 lb. weight of hammer. For hammers used in the ordinary way, or with the average of stoppages, the consumption may be placed at 13 cu. ft. per minute.

The cost of compressing to 90 lb., two-stage compression, is about 0.163 h.p. per cubic foot free air per minute, or for 26 cu. ft., as above, 4.24 h.p. and for 13 cu. ft. 2.12 h.p. per 100 lb. of hammer. To operate with steam under conditions similar to this last instance the same authority says that 1 h.p. of boiler capacity should be allowed for each 100 lb. of hammer, the boiler located within a reasonable distance.

The Star Drilling Company, Akron, Ohio, has three hammers with an aggregate weight of 4200 lb. At the lowest figure given above these would require $42 \times 13 = 546$ cu. ft., and the horsepower required would be $42 \times 2.12 = 89$. These hammers take care of six fires, and sometimes eight, and are supplied by two compressors whose nominal (greater than actual) free air capacity is 336 and 296 cu. ft. respectively, one-quarter of the air being used for other purposes. We have then $336 + 296$ cu. ft. $= 632 - 177$ (one-fourth) $= 455$ cu. ft., or 98 cu. ft., less than the lowest called for by the above rule. The compressors are driven by gas engines using natural gas; the power cost chargeable to the hammers is $52.28 per month.

Of the advantage in the use of air in this plant it is noted as follows:

"The hammers do more work than with steam. There is no trouble in starting to work out the water. Hammers work more lively and freely. No hot water dripping and no additional heat. The air-pressure is constant, while with steam there are serious fluctuations and the blows of the hammers are uneven. No steam is used in this plant and the installation is considered more economical and satisfactory with the air."

It is not here contended that it is in all cases advisable to use air for operating large hammers. The main blacksmith shop at the Phillipsburg, N. J., shops of the Ingersoll-Rand Company, the location and arrangement of which, for convenience, fficiencye and economy of operation, were given careful consideration, is very near the main boiler plant of the works, and the score of large hammers there are all operated by steam. There is in the blacksmith shop itself a Stirling water-tube boiler mounted in connection with the reverberatory furnaces to utilize the waste heat, this boiler being connected by an equalizing pipe with the main steam supply pipe of the works. When at times more steam is here generated than is being used by the hammers it goes into the main supply, while when, for instance, most of the hammers are working at once, the flow is the other way. Here as elsewhere the difficulty of ascertaining the actual consumption is apparent, but the arrangement in every other respect has been satisfactory.

In the oil tempering and tool dressing shop of the same works, located a little farther from the boilers, there is a hammer of medium size driven by air. This also is satisfactory, and the best arrangement under the conditions.

The air and steam consumption of steam hammers where any records are obtainable vary widely, as might be expected. The West Manufacturing Company, Buffalo, N. Y., tested a hammer 9 in. by 15 in. by actually running it continuously at 150 blows per minute, the air at 80 lb. gage, and using it at the rate of 230 cu. ft. per minute as measured by the piston displacement of the compressor. As a matter of fact, in actual, every day service, the compressor, running at the capacity above recorded, supplies this hammer, also a 7-in. by 12-in. hammer, five large air hoists, twelve small hoists, and a number of live air jets for blowing off scale, etc., and besides that the compressor "unloader," which stops the air compression, is in operation a considerable portion of the time.

The Buffalo Pitts Co. have a two-stage compressor with a nominal capacity of 350 cu. ft. of free air per minute, maintaining a pressure of 100 lb., which runs a 700-lb. hammer at 150 blows per minute. This compressor in regular shop work runs this hammer, also a smaller one, an air riveter and a number of smaller pneumatic tools.

The Shiffler Bridge Company, Homestead, Pa., have a two-stage, power-driven compressor with a maximum capacity of about 500 cu. ft. of free air per minute which drives two steam hammers, the air being used also for a great number of other purposes, the blacksmith shop with the hammers being a long distance away.

At the Painted Post shops of the Ingersoll-Rand Company there is a 500-lb. hammer operated by compressed air, and a

1000-lb. hammer, no longer in use, was so driven. Mr. F. W. Parsons, Superintendent, writes as follows:

"If the hammers were kept pretty busy doubtless steam would be the most economical of fuel, but, as they are often operated with long waits between, you get with the steam a good deal of condensation and this causes lots of bother when starting up, the iron cooling rapidly while waiting. Unless the packings are very well kept up water of condensation drips on the anvil and forging, which is annoying and sometimes dangerous. Of course with air there is none of this trouble, there is no loss when the hammer is not in operation, and with an equal pressure more work will be done using air than if steam were used. With long and exposed pipes, as are often used for steam to hammers, and especially where the hammers are not kept busy, it is more economical and certainly otherwise more satisfactory to use air."

The Elliott Frog & Switch Company, East St. Louis, Ill., are operating a hammer with compressed air because it is so far from the boilers, and they have an air supply for general purposes. They find that on account of the condensation of steam, etc., they operate the hammer to better advantage by the use of the air.

There are many railroad shops using air for steam hammers, but we have no specific information available concerning them. We never hear of the air being discarded after once being employed for this purpose, except for other reasons than those immediately pertinent to the hammer service.

At Ugine, Belgium, in connection with Girod electric steel furnaces, there is a large forge containing nine hammers operated by compressed air. The ram of the heaviest of these hammers weighs 10,000 lb., while the weights of the others range from 2000 to 200 lb.

The intermittent use of hammers makes a large air-receiver capacity desirable, and the heating of the air immediately before it enters the hammer is always promotive of economy The Trimvat Manufacturing Company of Boston have a 2000 lb. hammer which is supplied with air at 100 lb. by an electric-driven, two-stage air-compressor of 358 cu. ft. free air capacity, the unloader in this case being in operation about a quarter of the time. Large receiver capacity and highly efficient re-heating here exexemplify their advantages.

There are three air-receivers 54 in. in diameter by 12 ft. long, and the air is heated by twenty-four 1-in pipes, 2 ft. 6 in. long, connected to headers and placed in the hood over the forge fires. The heating in this case is so effective that the exhaust from the hammers is hot to the hand and has considerable unused pressure. The bills for current for this compressor and isolated hammer amount to about $140 per month.

It happens that this company has also a large steam operated plant which by the advantage of aggregation, and ignoring all the incidental advantages of air service, shows better economy for the steam when power cost alone is considered. There are one 5000-lb. hammer, six of 2000 lb. each and two of 1000 lb. The large hammer is used so infrequently that it is assumed to require no more steam than one of the 2000-lb. hammers, the aggregate to be operated, then, being thus equal to eight 2000 lb. hammers. These are supplied with steam by a 175-h.p. boiler. The coal, labor, etc., are estimated to amount to $35 per horsepower year, the total cost for the year then being $175 \times 35 =$ \$6125, and $6125 \div 8 = 765$ for each hammer, or $765 \div 12 =$ \$64 per month.

To use air instead of steam no change is required at the hammer, except that the exhaust pipe can be dispensed with. Pipe up otherwise as for steam and it is ready at once. Every one likes the air; it keeps the room cooler for the men; there is no water spattering on the hot forging and threatening to scald. Part of the exhaust can be used for blowing the scale off, giving cleaner and smoother work. The wear on the hammer is invariably in favor of the air, the wet steam, when used, washing away the oil, deranging the packing and causing leakage of piston, valves and stuffing boxes. With air the surfaces polish like glass, remain constantly more or less oily with little or no wear, and the hammer action is lively and prompt at all times.

There are several large concerns both in the United States and abroad now making steam hammers of the familiar types up to the largest sizes many of which are used for compressed-air service; these, however, it is not necessary to call attention to specifically. Full particulars concerning them may be found in the publications of the builders.

Compound-air Hammers.—There are, however, several compound pneumatic hammers which are worth considering. Generally speaking there is no excuse for compounding the cylinders of an air-operated engine or motor unless the air is passed through an efficient re-heater between the cylinders, just as there is no reason for two-stage air compression except for the intercooler between the stages. In the case of the compressed-air hammer, however, there is full warrant for it, inasmuch as it provides for the working of the air expansively instead of exhausting it at or near full pressure.

N. S. K. Air Hammer.—In Figs. 81, 82, 83, we have three vertical sections of the working parts of a, "N. S. K." compound

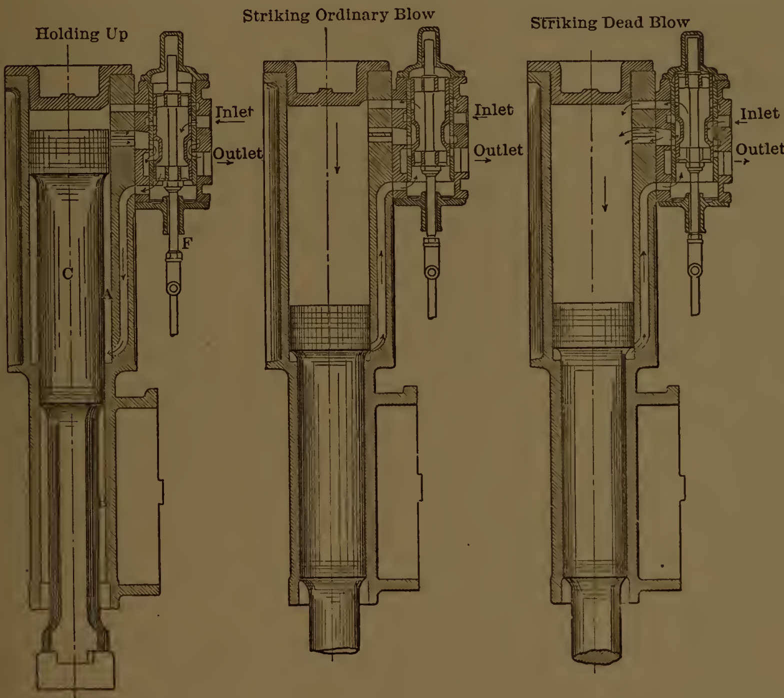

FIG. 81. FIG. 82. FIG. 83.

FIGS. 81–83.—Sections of N. S. K. Hammer.

air hammer built by Peter Pilkington, Limited, Bamber Bridge, England. Seventeen of these hammers, by the way, were turned out in one batch for Messrs. Harland & Wolff, Belfast.

There are two vertical tandem cylinders to the hammer, the lower one serving as a guide and being provided with a base by which it is attached to the hammer standard, while all the air work is really done in the upper cylinder. The differential piston provides an annular space A, Fig. 81, to which the air is admitted at full pressure for the upstroke of the piston. By the movement of the hand-lever-operated piston valve, after the upstroke is made, communication is opened between the annular space A, below the piston and the full area of the top of the piston. As the piston descends the air works expansively until at the termination of the stroke the air-pressure has fallen to very near that of the atmosphere, and all the pressure has been utilized on the way down. The movement of the valve for the admission of air to the annular space for the next upstroke also opens a passage for the exhaust of the air above the piston.

This cycle of operation is followed for the ordinary working of the hammer, and it results in a large saving of air over the use and discharge of it without the expansive working. When a very heavy blow is to be struck the valve is moved to its extreme limit of travel, as in Fig. 83, and then the air is admitted at full pressure to the top of the piston, giving great force to the blow.

The Massey Hammer.—Figs. 84, 85 and 86 show the principle and made of action of the Massey hammer, built by B. and S. Massey, Openshaw, England. This hammer uses the air expansively for all blows and minimizes the clearance losses. A further saving is in the prevention of the usual "after-flow" of the air into the upper end of the cylinder following the striking of the blow. This afterflow is induced by the piston traveling so fast in latter portion of the stroke that the air cannot follow quickly enough to maintain the pressure, and then after the blow is struck, the inflow of air to equalize the pressure is clearly a waste of air. In this hammer, this afterflow is prevented by the arrangement for working expansively.

Referring to the cuts, it will be seen that the lower end of the cylinder is always in free communication with the air supply through port A, the constant pressure serving to keep the piston normally in the raised position as in Fig. 84. The air inside the ram below the central guide or plunger is used as a spring or cushion. The combined effect of the two pressures—the constant one below the piston and the varying pressure in the middle of it—gives a resultant upward force which decreases as the ram approaches the top, with a cushion effect at the last, a rapid and lively action being thus produced. When the piston valve is raised, as in Fig. 85, the top of the cylinder is closed to the exhaust and put into communication with the bottom of the cylinder through the port B, and thus also with the air supply. The compressed air, therefore, passes from the supply on to the top side of the piston and drives the ram down.

As the ram descends the piston passes the port *B*, and in so doing cuts off the supply of air to the top of the cylinder. During the remainder of the stroke, therefore, the air already admitted is used expansively, and this expansion takes place no matter what kind of a blow is struck. For the striking of light blows with short stroke a small port can be opened by the attendant's

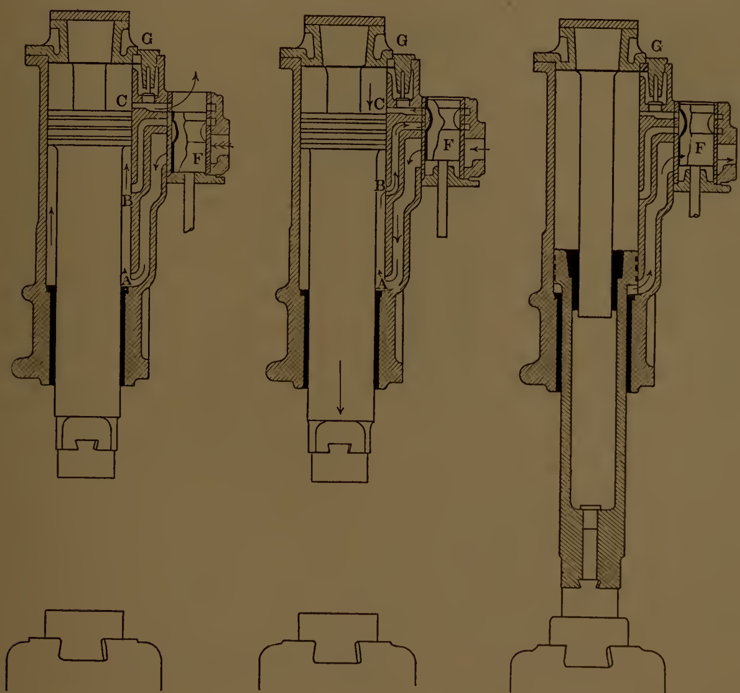

Fig. 84. Fig. 85. Fig. 86.

Figs. 84–86.—Sections of the Massey Hammer.

lever, admitting just air enough to allow the piston to descend. This port is closed again when the valves is moved further for a long stroke, while for holding-down purposes a second lever allows air at full pressure to flow into the cylinder even when the piston is at the bottom of the stroke. The small mushroom valve at *G* is intended for admitting the air-pressure into the space on top of the piston in case it has risen above port *C*.

The 5 cwt. hammer of this make has a ram 9 1/2 in. in diameter with a maximum stroke of 24 in. and weighs complete about 600 lb. It was operated with air at 45 lb. pressure with the following data as to power consumption, etc.:

Maximum: (*a*) Number of hardest blows per minute, 100; (*b*) cubic feet of air per minute for above, 310; (*c*) capacity of compressor for a single 5 cwt. hammer in cubic feet of free air per minute, 280; (*d*) approximate power required for working pressure of 45 lb.; 35 b.h.p.

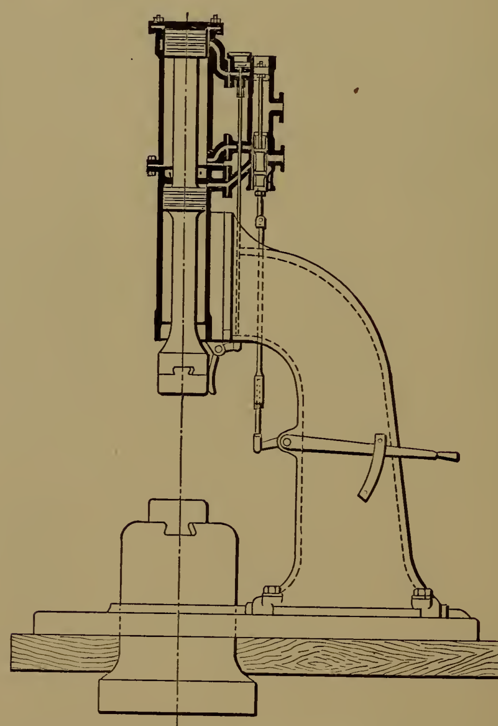

FIG. 87.—The Musker Hammer.

Average: (*c*) Cubic feet of free air per minute, 38; (*f*) approximate power required for pressure of 45 lb., 4.5 b.h.p. These powers are based on the assumption that the compressor requires for a pressure of 45 lb., 12.5 b.h.p. for 100 cu. ft. of free air per minute.

The Musker Hammer.—This hammer, as will be seen, Fig. 87 has two cylinders, the lower of which is operated only upon the down stroke, while the upper cylinder is, or may be, used for both strokes. The operating valve is actuated by the hand lever for each stroke as usual for steam hammers. Between this valve and the cylinder is what is called a "miter" valve, the function of which will appear. Assuming the tup or ram to be at the top and the valve down, as shown, to strike a light blow the valve is moved upward only far enough to admit the air above the lower piston, the same movement of the valve allowing the air under the upper piston to be discharged through the middle port to the lower exhaust.

To raise the ram the valve is lowered sufficiently to allow the air supply to enter the middle port, exhausting the air from above the lower piston through the lower port to the lower exhaust. The miter valve is opened by the exhausting of the air.

To give a heavy blow the valve is moved upward, allowing the air to go through the upper port, under the miter valve, and to act upon the upper side of the upper piston and at the same time through the center of the valve of the lower part, and to act upon the top of the lower piston, the air then acting downward upon both pistons at the same time.

For the upstroke the valve is reversed; air-pressure is admitted underneath the upper piston through the middle port, and the air above both pistons is exhausted through the upper and the lower ports respectively.

The miter valve serves for using the air expansively when striking heavy blows. It is arranged to be kept open by a curved lever arm resting against the ram until the latter descends about half-stroke, when the arm is freed and the valve descends, cutting off the air-pressure, the stroke being completed by the expansion of the air already in the cylinder.

Economy of working results from both the using of the lower piston only for light blows and from the expansive use of the air when the upper piston with the large area is used. A comparative test of air hammers was made by Sir William Armstrong, Whitworth & Co., Limited, with the result, as stated, that the Musker hammers used less than half the air required for others.

CHAPTER XXVI

DIVING BELL AND CAISSON

An inverted glass tumbler lowered into a pail of water, the air in the tumbler restraining the water from rising within and filling it, is typical of one of the most important lines of devices in which compressed air is employed as the responsible agent. The principle of water exclusion by air-pressure is so widely applied that it is not easy to determine which application of it should come first to be spoken of.

It has for instance, been frequently employed for theatrical effects, notably at the Hippodrome, New York City, where a large water tank is a prominent and permanent feature of the stage paraphernalia. Perhaps one of the most striking of the effects produced, and one most easily explained, was where a considerable body of men, and also of women, went four abreast in steady march time down a stairway into the water until their heads were submerged and they disappeared entirely not to be seen again.

The sketch here given, Fig. 89, is purely imaginary and is made by the writer without any knowledge of the actual details. *A* is the surface of the water in the tank and *B* is a fixed box without bottom constituting the exact counterpart of the inverted tumbler. The water does not rise within this box because it is kept filled with air at a pressure of not more than 2 or 3 lb. to the square inch, sufficient to keep the water out. Immediately before the amphibian marching act the surface of the water in the tank was agitated and large and numerous air bubbles began rising, these showing that the chamber *B* was full of air which had expelled the water and that some excess of air was escaping under the lower edge. When the march was on, and the first row had just reached the lower step and their heads had just disappeared below the surface of the water, they had only to duck their heads to pass under the edge of the chamber and then when they stood upright with their heads within the chamber they would at once begin to breathe again. Then the chamber being continued as a

FIG. 88.—Section of Tunnel Caisson—River Seine, Paris.

narrow passage to the side of the stage they could turn at once and march single file with their heads out of water and be out of the way of the row of marchers following them. The passage extending beyond the visible side of the stage then steps could lead the marchers up and out of the water, and an air lock would release them from the air-pressure. The pressure being, as was said, not more than 3 lb., the marchers would not experience serious inconvenience on that account. The compressor employed would not require to be anything more than a foundry pressure blower, and the air lock might be simply a revolving door, such as we are familiar with in office buildings.

The diving bell which is the inverted tumbler upon a large scale, and like it movable in any direction, is now seldom used or

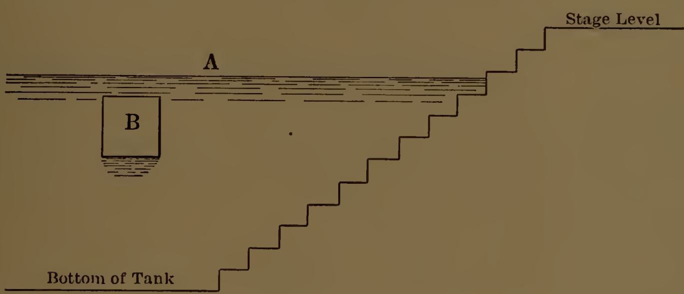

Fig. 89.—For the Submarine March.

even thought of, and in its day it did very little notable service. It could be lowered over anything not too deeply submerged and the occupant free to breathe could reach down into the water and do things, but the limitations of work in such conditions are evident.

From the diving bell to the caisson is but a step so far as the principle involved, but the latter has become the responsible and effective device relied upon for the execution of the most important engineering works. It has become apparently a necessity for the construction of bridge piers and abutments, for lighthouses, for dams and locks or for any structures whose foundations must be laid below the water level. The caisson so employed differs from the diving bell in that when once put into use it is not movable, except vertically downward as the work proceeds, and that when the required depth is attained, by reaching rock or reliable bearing strata, it remains there to be filled with solid material and becomes a permanent part of the structure to be erected.

The caisson is now perhaps more frequently employed in what in the beginning would seem to be strictly terra firma operations. The buildings which modern enterprise finds it profitable to erect reach to great heights and are of so great weight that the foundations must go to great depths to find unyielding support. In lower New York City especially, but also in many other places, it happens that before the required depth is reached the water begins to force its way in, and caissons are as much required here for the exclusion of the water as in strictly subaqueous operations. Practically all the skyscrapers of lower New York City have pneumatic caissons for their foundations, and the "sand hogs" who find special employment in the sinking of the caissons have become sufficiently numerous to boast of their occupations as a distinct trade, with rights and regulations and requirements of its own. Thanks to the caisson, the erections of the last quarter of a century in our modern cities may boast a permanent stability at the base such as is revealing itself as wofully lacking in some of the most celebrated ancient and medieval structures of the Old World.

When the inverted tumbler is first lowered, and when the tip of it just touches the surface of the water, the air is at once prevented from escaping, but the volume and pressure of the air are as yet unchanged. As soon as the tumbler is lowered at all the water rises more or less within it, there is at once a certain increase of pressure in the air and its volume is reduced. The pressure to which the air is subjected under these conditions is the sum of the normal atmospheric pressure at the time and the pressure due to the difference in the height of the water within the tumbler and the height of the surface of the water outside.

In the larger practical tumbler, or caisson, each foot of difference in the height of the water outside the caisson and inside is equal to 0.4466 lb. per square inch. This is the pressure which would be shown upon a gage exposed to the pressure of the atmosphere and connected by a tube or otherwise with the interior of the caisson. In a caisson more or less submerged the abnormal pressure to which the workers in the caisson would be exposed would be that due to the water height outside it. If this height were 50 ft. then the pressure within the caisson would be $50 \times 0.4466 = 22.33$ lb., gage.

In practice the water does not rise in the caisson, the pressure within being assumed to be sufficient to keep the water down. If the pressure of the air were not artificially increased by the forcing in of more air the water would rise to quite a height within the caisson. It would require special computation to ascertain the height to which the water would thus rise within the caisson, but this never occurs in practice, and this is where the work of the air-compressor begins. Sufficient air-pressure is maintained within the caisson to prevent the water from rising within it at all, and this pressure will be that due to the outside water height. There is no care required at the compressor for the nice adjustment of this pressure; it being only necessary to keep the compressor constantly working and to send down an excess of air, the surplus escaping under the lip of the caisson, so that the pressure required to balance the water pressure cannot be exceeded in any case.

This excess of air supply is a constant necessity, as the air in which the men work becomes vitiated by respiration and otherwise and would become unbreathable if not renewed. It is necessary to take means for keeping the air sent through the compressor as pure as possible to begin with, and as free as possible from oil vapor and other impurities which the compressor might contribute, and it is also necessary to deliver the air at or near normal temperature, the general effect of the compression being to heat the air, so that for caisson and tunnel work an after-cooler for the air is extremely desirable, if not imperative.

The pressure of the air at the bottom of the caisson has a lifting effect upon the entire horizontal area, so that provision must be made for weighting the caisson to force it down. As the caisson, when sunk to its permanent position at the required depth, is to be filled entirely with concrete to form a solid mass to sustain the steel columns or other superstructure, as much of this permament load as possible is placed upon it during the process of sinking. At the bottom of the caisson is left a chamber of sufficient height above the lip for the men to work in, this being covered by a deck with a sufficient central opening for men and material to pass up and down, this opening or vertical passage being extended upward as far as necessary and closed with an air lock for retaining the air-pressure. Then upon this deck and around the central passage while the sinking is going on and the men are working in the pressure chamber, there is built up the solid mass of concrete or masonry which is to form the permanent structure, this being carried up to a considerable height above the ground. In addition to this permanent load large weights of cast iron are often piled on top to help the sinking. Besides the actual air pressure below there is often a large earth friction around the sides of the caisson to be overcome. When the lip of the caisson has reached the solid rock, or whatever is accepted for the permanent bearing, the working chamber below is filled up solid and also the working shaft all the way up, the air locks and piping being removed. While the filling up is going on the air-pressure is gradually reduced or altogether withdrawn when no longer required, and the permanent load rises upon the completed foundation.

The depths to which pneumatic caissons may be sunk is limited by the ability of the men to endure the air-pressure, and it is a rather curious fact that the depths below the surface at which solid rock is found in New York City approach very nearly without exceeding this limit. In sinking the foundation caissons for the city hall the maximum pressure worked in was 45 lbs., which is very near the limit record for this kind of work.

There has been much planning for a bridge across the North River at New York, such a bridge requiring one or more piers in midstream, but it is considered that pneumatic caissons would be impossible on account of the depth and the consequent air-pressure involved, and so that scheme waits for some other device which would be practicable and not too costly.

The Caisson for Subaqueous Tunnel Construction.—The two halftones (Figs. 88 and 90), help to tell the story of another interesting type of caisson work. The lines of urban transit in Paris cross the Seine five times upon bridges of various design, but one line, traversing the central portion of the city, passes under the river and in its construction a thin ribbed mode of construction was employed. The plan first proposed was to have two tunnels driven at a suitable depth by the compressed-air shield method. This would have necessitated the adoption of a level at least 10 ft. lower than was required for the system adopted while at the same time the latter would permit the use of a single tube for both tracks instead of a separate track for each.

At each side of the river and running out to a certain distance from the banks the tunnel was built or driven by the aid of a compressed-air shield and then caissons were sunk in the bed of the river to the required depth and in correct positions to form a continuous tube. There were three of these tunnel caisson

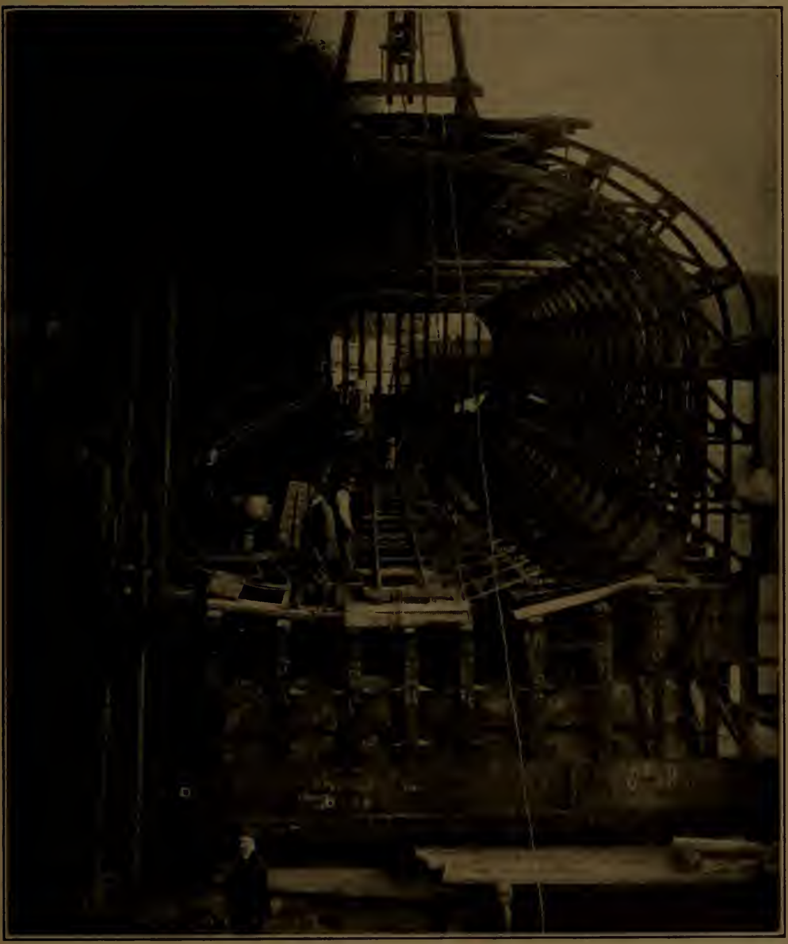

FIG. 90.—End View of Tunnel Caisson.

sections in the large arm of the river and two sections in the narrower arm.

The halftones show quite clearly the construction of one of these caissons, some idea of the size of which may be formed by comparing with the men employed around it. The structure comprised two principal parts, the tunnel proper and an outer framework which formed the actual caisson. The tunnel portion was approximately elliptical in section and composed of iron rings. Each ring was made up of a series of cast-iron voussoirs bolted together and the rings then bolted to each other with layers of treated wood between for a water-tight packing.

Around this elliptical tube was built a metallic shell which in the lower part of it formed the rectangular caisson. The frame of this shell was formed of a series of curved ribs of channel iron outside the tunnel tube and braced at a certain uniform distance from and nearly parallel with it by cross-braced ironwork. These ribs did not run around the under side of the elliptical tube, but, passing the center or horizontal axis, they extended vertically downward thus forming the sides of the caisson and extending some distance below the bottom of the tunnel they thus formed the compressed-air chamber in which the men worked to make the necessary excavation which allowed the whole structure to sink to the required depth. Over the ribs was placed a sheet iron covering which formed an external air-tight shell, the ends also of the caisson being thus closed.

Each caisson was floated and towed to its place, piles having been driven there to guide it vertically during the sinking operation. The semi-elliptical space between the tunnel tube and the sheet-iron sides, all the central horizontal plane, was filled with cement baton, the top plating having been left off for the purpose. As the ground was excavated within and under the caisson by the "sand-hogs" it was gradually lowered to its proper level below the river bed, the tunnel tube having been ballasted with water to give the needed weight and steadiness. When finally in place the caisson below was filled solid with concrete, the shafts at the side, with the air locks for the passage of men and material, were removed, and after all the connections were made the water was pumped out of the tunnel tube. The ends of the caissons lay about five feet apart, and this space was filled by a small caisson which was sunk last. The removal of the end plates of all the sections left the continuous tube clear.

There was a double pneumatic interest in this undertaking, as compressed air was employed not only in the sinking of the caissons but also in their construction, both of the halftones showing pneumatic riveters in use upon the frames and shells.

The Submarine Diver.—The so-called "armor" worn by the individual diver, giving freedom of movement under water, for doing all kinds of submarine work is still another application of the inverted-tumbler principle, although the original shape of the device is entirely lost sight of. The diver is clothed in a water-tight suit and then air is forced into the helmet portion at sufficient pressure to balance that of the water. In this case also no attention is required as to the maintenance of the proper pressure, only a sufficient and constant influx of air being required, the excess escaping and the pressure within automatically adjusting itself to the pressure without. The inflow and the escape of the air go on continuously, so that the diver will have fresh air to breathe. If the diver changes his position vertically the rate of escape of the air will vary accordingly, being more rapid when he ascends and less rapid when he descends, all his movements naturally being slow and deliberate.

The diver's armor as now used has developed many refinements promotive of safety, of comfort and of facility and rapidity in working. The electric light and the telephone being now both available to the diver, he can do many things which his predecessors could never have thought of doing.

Raising Ships

The bouyancy of the inverted tumbler suggests the use of air for raising ships and it has often been successfully employed for this purpose when vessels have been at workable depths. A ship's hull floats because it has large empty spaces and when these fill with water she inevitably sinks, but if the water can be expelled she will rise again. When in a submerged vessel the sides and the deck are air-tight or can be made so by divers, if air can be forced in in sufficient quantity to expel the water, the vessel must rise no matter what may be the condition of the bottom. A most necessary precaution in such a case, and often a difficult one, must be to see that the entire ship rises together. It will not do to have the air accumulate and create a great preponderance of buoyancy at either end. There must be some central partition or other means of distributing the air equitably to each end so that both ends will rise. When once afloat on even keel, there is no longer much difficulty in keeping a vessel righted and towing her to dock or elsewhere.

Although compressed air has recorded a number of notable successes in the raising of sunken ships, its opportunities in that field are necessarily few and infrequent from the fact that so so few ships sink in such places or at such depths as to make it possible for the air to do anything in the matter

A vastly greater work for compressed air lies before it in the preventing of ships from sinking. Here its possibilities apply not to a very few but to many or to all ships. If the unsinkable ship is ever built, or if such a consummation is ever nearly approached, it will never be without the employment of compressed air. Under compulsion increasing attention is being given to this matter. The sinking of the Titanic will never be forgotten until the repetition of such a catastrophe has been made impossible.

The most important device for keeping modern ships afloat is the transverse bulkhead, and it is a feature of all ships now built. That the bulkhead has actually saved many ships is matter of record. Typical local examples, as we might call them, we carry in our memory, all of them on trans-Atlantic lines. There was the Arizona in 1879, the "Greyhound" of the Atlantic of that period; she struck an iceberg in November, not the iceberg season; her first bulkhead held and she was safe. Then a few years later the City of Berlin had a very similar mishap—if you call it a mishap—of ramming an iceberg in the season of ice. Her bow was smashed into a shapeless mass, as the writer remembers the sight of it, but her forward bulkhead saved her. In 1889 the City of Paris had a novel accident, her engine having punched a hole through the bottom, and she was saved by her after bulkhead. When her engine-room was flooded and the ship was pitching, and the water was swashing back and forth, there was an anxious time of watching, and questioning as to whether or not the life barrier would hold, and only after this had been hastily but effectually braced by timber on the other side was the anxiety relieved.

Bulkheads have not always held in times of great stress, and disaster has resulted. It is the practice to fill the sections successively with water to see that they are watertight before the ship is launched, but this gives no sufficient assurance of strength, and it is not so easy to devise a test which should satisfy.

While the collision bulkhead is one of the simplest of devices as to function, not a thing apparently which any engineer should need to study over except as to strength required and other details of construction, still it is a fact that even this simple thing has been misapplied, and it may also be made to appear that the application of it which might be most valuable of all has never yet been made.

Of the unwisdom of great engineers over a very simple thing there is the episode of the longitudinal bulkhead in evidence. We can see now easily enough how the longitudinal bulkhead works. If a rent is made in the side of a ship provided with a central longitudinal bulkhead, and if one side of the ship, or even a not very large portion of one side, fills with water while the other side is not filled the ship of course rolls over. Yet, the British Admiralty had to lose at least two great ironclads, the Captain and the Victoria, with a loss in those two of perhaps as many men as went down with the Titanic, before this was realized.

The bulkheads upon which we rely so much for the saving of our ships from sinking—when they save them—are water-tight transverse partitions which divide the interior of the hull into sections with the result that if any one, or any two, or sometimes if any three of these sections fill with water as the result of a collision, or the striking of a rock or an iceberg, the ship will still keep afloat. It is understood by the writer that in the Titanic there were fifteen of these bulkheads, or sixteen separated sections. Any two adjacent bulkheads exercise their saving function not at all upon the section which they enclose between them but on the section adjoining this one on either side. The enclosed section may fill as it will and the bulkheads offer not the slightest hint of opposition. Yet why not?

Indeed, to protect the section which any two bulkheads jointly enclose should be their most important and imperative service. If each section besides its two watertight, or, for our present purpose, airtight bulkheads, had also an airtight deck enclosing it above, with airtight hatches or doors, and if absolutely nothing more were done or provided, then if a hole however big were stove into or near the bottom so that the water would rush in the section *could not be more than half filled* with water. The air which normally filled the section at a pressure of 1 atmosphere could only be compressed to a pressure, as long as the ship floated, not exceeding 2 atmospheres, or 15 lb. to the square inch by a pressure gage, and to a volume not less than one-half of what it was originally, and there the water would have to stop.

If now compressed air were forced into that half-filled section at the pressure required, not exceeding 15 lb. gage, the water would all be driven down and out, that is down to the top of the hole by which the water had been rushing in, and to then hold the water from again rising above that level it would only be necessary to maintain the air-pressure by forcing in just enough air to make up for whatever air leakage might occur. It would not be necessary to wait for the water to rise in the section before beginning to force in the air. This should be begun as soon as the water began to rush in, or as soon thereafter as possible. No care would be required about the air except to supply it in sufficient quantity. Any excess of air would be driven out at the bottom with the water and no increase of pressure could be thrown back upon the compressor. This is how the air is disposed of when men are working in foundation caissons. The air must be continuously renewed so that it may continue breathable, but it is only necessary to continue forcing the air in; its discharge will take care of itself.

Any compartment of the Titanic between two bulkheads and beneath the lower deck may be assumed to have had a cubical content something like this: 90 ft., abeam; 50 ft., fore and aft; 30 ft., deep = 135,000 cu. ft. A considerable portion of this space would be occupied by coal or cargo, so that we may assume a space of 100,000 cu. ft. to be filled, or, in addition to the normal atmospheric content of the compartment, there would be an equal additional volume of air to be supplied, and with air compressors having a free air capacity of 5000 cu. ft. per minute, the space would be filled and the water would be expelled in 20 minutes, and the maximum power required, would be 250 h.p. It is not necessary here to go into the practical details of the arrangement suggested, but it would be well to have oil-engine-driven compressors, which can be started at a minute's notice, located upon an upper deck, preferably one at each end of the ship and connected to deliver into the same pipe system, with all necessary valves, etc.

It would seem to be a somewhat strange and unaccountable thing that there could at this day exist an opportunity for the writer, or for any one else, to be calling attention to this matter.

There is nothing in the way of originality of invention in the suggestion, no special ingenuity is required in the application of it and there can be no question as to the results which would be secured.

If even now the providing of watertight bulkheads is made compulsory under the law, and very properly so (although it should provide more adequately for having them strong and stiff enough) it would seem that the providing of airtight decks and hatches, so that any section may be made a pressure chamber and water-excluding, should be equally so.

Although this suggested arrangement promises really an additional element of safety greater than that contributed by the watertight bulkhead itself when not assisted by the compressed-air feature, it entails little additional expense. The deck must be there in any case, and such decks even now are practically airtight. Some attention should be given to the bracing or strengthening of them to withstand the upward pressure, and there would be the providing of the airtight hatches, and in some cases double doors or airlocks would be needed, and that is all. The latter suggestion recognizes the fact that it would be easily possible for men to work in the air thus employed to hold the water back, for making repairs or stopping leaks from the inside, for passing up supplies, etc., the pressure being much less than that in which men work in sinking foundation caissons or in driving subaqueous tunnels.

Thus far our thought has been of ships in general, with the passenger ship most prominent; but ships of war are really most ready of all, and also most in need of the application. They have the bulkheads and they have the decks with few and small openings through them, all of which may easily be provided with means of prompt and ready closure, and then all that is needed is the air compressor.

The arrangement of the airtight bulkheads and the airtight deck would not generally be sufficient to keep afloat a warship of the modern type without a compressor to expel the water from all the space enclosed on account of the low freeboard and the limited space enclosed.

In connection with warships especially it is proper to mention another service which compressed air might render in some emergencies. By the controlling of the air pressure in the different sections and a proper manipulation of the bulkhead doors masses of water might be easily directed from one section to another for the purpose of keeping the ship on an even keel, thus serving to prevent or defer the settling endwise which usually hastens, if it be not the actual cause of a final catastrophe.

The following, mostly abstracted from *The Engineer*, London, tells us of what is actually being done in the direction above indicated. The U. S. battleship Pennsylvania, also the Nevada and the Oklahoma, are being equipped with an installation for localizing and minimizing underwater injuries, under the direction of Mr. W. W. Wotherspoon, who has had successful experience in the raising of sunken ships by the aid of compressed air.

Every man-of-war is subdivided, from a point above the water-line to her keelsons, into many hundred separate water-tight compartments, for the purpose of confining and localizing injuries. These divisions are connected to an extensive drainage system, and ordinarily powerful pumps are counted on to overcome leakage. If a compartment filled, notwithstanding the pumps, and the invasion stopped there, the consequences might not be very serious; but, unfortunately, the pressure upon bulkheads and decks is often too great; hence they yield, and the neighboring compartments are then flooded, and soon, the ship slowly but surely careening or sinking.

Mr. Wotherspoon subdivides a ship into successive strata or layers of compressed air zones, and he distributes these as the emergency arises so as to meet the requirements of each exigency. In other words, the pressure of the air on the opposing sides of decks or bulkheads is such that the resultant difference on the side of greater pressure is a number of pounds less per square inch than would be the case if the compartment were filled with water and flanked only by normal atmospheric conditions. In this manner the structure of the damaged space is supported by the enveloping body of the ship throughout a pretty wide area, and this effectually guards against the yielding of the walls directly encompassing the damaged region.

Mr. Wotherspoon having met the official objections which were raised, the armored cruiser North Carolina was allowed for a test installation. It was elected to connect the apparatus with something like 800 of the ship's water-tight compartments, narrowly limiting the weight of the devices employed.

The underwater compartments of all men-of-war are forcibly ventilated to guard against the accumulation of gases and foul air. This is effected by a double system of piping, one carrying fresh air into the compartment and the other supplying an outlet for the tainted exhaust. Running down below the water line as it does, all of this piping is subjected to pressure tests and must be equal to any tax which may be placed upon it by water pressure due to flooding, and therefore these conduits already in place stood ready to serve another purpose. By these channels Mr. Wotherspoon can lead compressed air to any of the compartments, and the only additional apparatus needed for this work is a flexible attachment for effecting connection with a source of air supply. Originally this was the air compressors, but it is now intended to have air stored in reserve in sufficient quantity to help toward the immediate checking of any dangerous admission of water.

A supplemental feature is that of air locks extending from the uppermost water-tight deck down to the lowest water-tight spaces, and these air locks, by means of suitable vertical doors—not the top doors—on each deck make these convenient passageways between decks for all ordinary traffic.

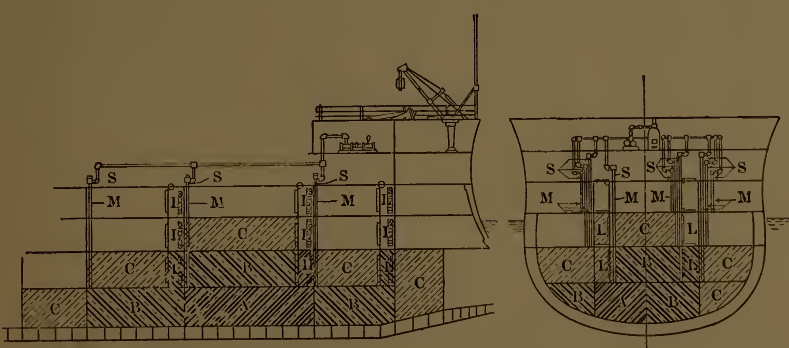

Fig. 91.—Compressed Air Chambers on Warships.

The athwartship section, Fig. 91, shows the general arrangement of the system upon one of the United States naval vessels. Compartment A is the damaged one, and theoretically is under an air pressure of 14 lb. to the square inch. The contiguous compartments B B B are filled with air at a pressure of 9 lb. per square inch, and this leaves a resultant bursting pressure upon the walls of compartment A of only 5 lb. per square inch. The outlying spaces C C C are charged with air at a pressure of 4 lb.

per square inch, and again this leaves a resultant bursting pressure of 5 lb. within compartments B B B. Therefore instead of the walls of the damaged compartment A having to withstand the entire stresses due to the confined air at 14 lb. pressure per square inch, the flanking and superposed enveloping ship fabric is bearing its part progressively.

According to the building specifications of every fighting craft all of the water-tight compartments are required to be tested some time during the ship's construction. This test for water-tightness when carried out does not involve pressure testing to any pronounced extent, and the spaces are seldom, if ever, filled as they would in all probability be in case of flooding due to damage. In the case of the North Carolina it was found that more than one compartment was leaky when the compressed air was used. Not only was this so, but a good many water-tight doors theoretically perfect proved to be anything but tight. In most cases this was found to be due to worn gaskets and loosened fittings, but these potential leaks would have escaped notice but for the telltale hissing of the liberated air. Thus, apart from its virtue in case of injury to the bottom platings, the system as installed upon the North Carolina showed how valuable it could be in checking the water-tightness of the craft within its own body structure. The secondary usefulness might easily resolve itself into one of prime importance under some conditions.

Mr. Wotherspoon has made his system valuable for the smothering of fires as well as for the arresting of leakage. Instead of turning compressed air into a compartment he chokes out a conflagration by using the same conduits to carry a gas which will not support combustion into the endangered division.

A third advantage of this dual fire-and-water protecting system is the peculiar facility which it offers for making temporary repairs. By means of the air locks it is possible for the men to enter an affected compartment and to overhaul a leaky valve and replace it without any inconvenience. Again, should the injury involve a rupture of the ship's inner or outer plating, that, too, can be temporarily stopped or plugged or otherwise sealed as the occasion may best afford. The men would simply be working in the structural double of a caisson.

CHAPTER XVII

AIR JET—SAND BLAST—CEMENT GUN

First it will be worth while to note what is done by the air blast alone. Dry, clean air, bearing no sand or other material is used for polishing small articles of metal, in this case the high velocity of the blast impinging upon the metal producing the polishing effect. In the centrifugal machine the load, whatever it may be is normally accompanied at the same speed by the air which fills the interstices. With the basket of the machine rotating at its maximum speed a jet of air is pointed to blow in the reverse direction, and this is found to have a polishing effect upon the contents. Nickel-plated articles, for instance which have become tarnished, and small articles generally which require a finishing touch are brightened up very quickly, and this has become an established practice in certain factories. It is only necessary to pack the articles, which are generally small, so that the air can attack them all over. It may not have been suggested, but it might be well without changing the loading to reverse both the direction of rotation and the pointing of the air jet. This would give the air a surer chance of getting at all sides of the articles to be polished.

The Air Blast in Foundation Sinking.—A contracting company in New York City is making a specialty of the use of the air blast as an aid in sinking building foundations. In this work the foundations are sunk to rock which is overlaid by varying depths of gravel or other compacted material. A recent undertaking was the construction of the foundations for a large twelve story building where the rock was below the surface at depths ranging from 17 to 50 ft. When completed the foundation consists of a series of reinforced concrete structures, these being formed of groups of 12-in. steel pipe filled with concrete among which are distributed vertical reinforcing rods of steel.

The pipes used are in lengths of 12 or 14 ft. and in the deepest parts three or four of these are required, one above the other. Each pipe after being properly located in a vertical position is driven down a few feet by an air-operated pile-driving hammer placed upon the top of it.

The pipe having been thus driven a certain distance the hammer is lifted off by a derrick, the pipe is filled with water, a compressed-air pipe is thrust down as far as it will go and also a test rod which is used to ascertain when the solid rock is reached.

On the street close to the work is located a 20-in. steam actuated air-compressor and a large receiver. The compressor is speeded up and when the receiver is filled to the highest working pressure all the air is suddenly turned into the foundation pipe, blowing out the water, the dirt and the stones. This filling with water and blowing out is repeated two or three times if necessary until the air pipe has been sunk to the bottom of the casing pipe. Then the hammer is put to work again, the pipe is driven down another couple of feet or so, then it is filled with water and blown out again until the rock is reached. The pipe is even finally driven a foot or more into the rock, the familiar New York mica-schist.

When the driving for the individual pipe is finished several steel reenforcing rods are inserted and then both the enclosing pipe and the rods are cut off to the required level by the oxy-acetylene torch, the pipe is filled with concrete and finally capped.

This method is also being successfully employed for putting in new foundations under old buildings, a section of the building being cut out to give room for a short length of pipe and a hydraulic jack, the pipe being alternately jacked down and blown out and followed by additional lengths of pipe until the required depth is reached.

The Sand Blast.—In the case of the sand blast the air jet is made to carry an abrading material and to deliver it with such force that the material has a sharp effect upon the surfaces by which it is arrested. It is not easy to name any specific employment of the sand blast which may be called its principal work. For the cleaning of castings it may be said to be used in every enterprising and up to date foundry, so that in this line it is not only numerously employed but also the air is used in considerable volume in each case.

In the progressive extension of the use of the sand blast the tendency, as in similar cases, has been toward simplification of apparatus, some of the changes resulting perhaps in an apparent sacrifice of ecomony in the use of the air. The use of a high pressure for the air rather than a lower, or *vice versa*, can scarcely be regarded as a simplification one way or the other. There is still more or less discussion as to whether low pressure—15 or 20 lb.—or high pressure—80 to 90 lb.—is to be preferred.

The low-pressure advocates recommend the use of a pressure reducer and an auxiliary air-receiver near the work when the supply is taken from a high-pressure service. The low-pressure system will use larger nozzles and cover a greater area of work at a time, but the sand will not impinge with as great force as with the high pressure.

Both the sand and the air must be dry, the latter, of course, not absolutely so, and some knowledge of the means of securing air comparatively dry will be likely to leak into the mind of any one who becomes at all familiar with the present volume. Sand in foundry practice is used over and over, and if it becomes damp so that it will not flow with perfect freedom it will require to be dried in an oven or otherwise. There are various effective sand driers in the market.

Sand should be clean and sharp and for the larger nozzles it need not be as fine, and indeed is better not to be as fine as for the smaller nozzles. The sand does all its work by the suddenness of its impact upon the metal, so that it is more or less shattered into smaller fragments or dust, which becomes useless and must be sifted. Sand for the sand blast is being manufactured in different grades by the crushing of suitable rock and sifting to the required sizes.

Different pressures are recommended for different kinds of work, the requirements for ordinary castings ranging from 5 to 20 lb., while for steel castings as high as 75 lb. is recommended.

The sand blast is used with various arrangements looking to expediting the work or protecting the workman. Fixed jets are often used to deliver the sand blast directly upon the contents of revolving tumbling barrels. A slowly but continuously revolving turntable has been employed with a heavy leather curtain suspended across the center, one-half of the table being exposed to several fixed blast nozzles which will effectively clean the castings and as these emerged from behind the curtain they can be successively removed and replaced by others.

In one case the sand blast is on one side of a tight partition and the workman on the other side reaching his glove-protected hands through holes in the partition for directing the blast or manipulating the castings and looking through panes of glass placed at a convenient height. In the place where this arrangement was in use it was necessary to replace the panes of glass every day.

The rapid wear of the nozzles is the most troublesome thing about the sand blast. The hardest steel needs frequent renewing being finally discarded because the bore wears too large. The best thing known to the writer is so-called "white" iron, or the iron of which malleable iron castings are made but without the "annealing," or rather the carbonizing process. These castings are so hard that they cannot be machined or even ground at all, and must be used just as they come from the sand. They cannot be cored much less than 3/8 in., but we have seen many such used for rough work, say about 1 in. in diameter outside and 6 in. long. These are easily connected to a rubber hose.

The sand blast is frequently used for cleaning the outsides of stone and other buildings, and as only light pressures are required, varying with the height at which the work is carried on, the air can be supplied in sufficient volume by an electric-driven or a gasoline-driven portable compressor, and apparatus of this type is not unfamiliar in the large cities.

The following concerning the use of the sand blast in gold and silver art work is a brief abstract of a portion of a valuable paper by Frank Mason of the University of Sheffield. It is not only intrinsically interesting, but it is more than that, suggestive of the many other possible uses of the sand blast for the delicate manipulation of surfaces of wood, glass, metals, etc.

The continued demand for variety in the finish and appearance of gold, silver, and electro-plated goods, gives to the sand-blast such a wide field of operations as to be almost unlimited, and the opportunities, now opening up, for its application are without parallel in its history. As a ready means of imparting to silver and other materials, at a small cost, a finish very pleasing to the eye, it is hard to rival. From the number of machines specially built for different classes of work, constantly being placed on the market, it would seem that blasting-machine manufacturers are alive to the possibilities of their productions. These are now so numerous that to select the most suitable would be almost impossible without first being acquainted with the class of work to be done. The two chief governing factors of the process are the pressure and the material used. Obviously, therefore, the method of procedure may be varied in two directions.

1. By varying the pressure used in forcing the blasting material against the article.

2. By using material with varying cutting properties and of different grain.

A matte may be obtained in the former case, very deep or slight, but not necessarily coarse, even under a pressure of 18 to 20 lb. per square inch. This factor (the power), then governs to a large extent the depth of the matte and varies between about 2 to 20 lb. per square inch, according to class of work and desired ultimate appearance. The question of blasting material is one of undoubted importance and yet, strange to say, is very often overlooked. A visit to a continental plate and jewelry establishment would very quickly convince the sand blaster of the obvious difficulties he would encounter in endeavoring to produce such surfaces as he would see displayed in a first-class house. The peculiarly frosted, delicate, French gray finish so dear to the artistic Parisienne, and to be seen all over the continent is a striking example of the careful manipulation and the delicate treatment necessary in some sand-blasting operations. In this class of work this process is the last to which the article is submitted, hence its requirement of very careful treatment, judicious selection of blasting material and well-regulated low pressure.

Another instance in which very careful manipulation is required is in the production of the lovely, soft, greenish gold tint so much admired. This is obtained on gold-plated articles, as well as those manufactured wholly from the metal itself. Here again under even a moderate pressure from the sand blast, the deposit of gold might be completely spoiled. The method adopted in most cases to obtain the above finish is first to gild or gold-plate the articles, bringing them from the gold bath a little darker in color than is required in the finished appearance. If heavily coated, scratch brush, preferably with very fine German silver wire brush, and blast with flour of pumice under a pressure not exceeding 3 lb. per square inch. Wash away any pumice clinging to the article and wipe with very soft chamois leather.

A very fine surface appearance, usually employed on articles having parts in high relief, is obtained by means of the sand blast on "oxidized" silver as follows: The process of sand blasting should be operated just prior to silver-plating, using powdered pumice and a pressure of about 12 lb. After completing the deposit, "oxidize" same in a solution of potassium sulphide until the article assumes a rich deep blue, or blue back color. Then by means of a calico mop or dolly and Trent sand, relieve where necessary according to design. The result of this process is quite a study in light and shade and is productive of some very fine effects. Satin finish, now often produced by means of the sand blast, is a term used to indicate the appearance of articles bearing a matted surface.

Crystalline, ice-like surfaces similar to some molded glass wares, are sometimes desired. It is necessary in these cases to employ a high pressure, say 18 lb. per square inch. As blasting material, a coarsely ground glass or very coarse sand should be used. This, of course, is done before plating, and as it is quick and severe in operation must not be overdone by prolonged blasting. "Partial frosting," as the term suggests, is the production on one surface of a combination of matted and brilliantly burnished parts.

This gives to the article a very pleasing contrast and is imparted thereto by means of the stencil. It may be accomplished very readily by cutting from ordinary writing paper or similar material the pattern or parts to be left bright or burnished, and then glue same to the article. Allow to set thoroughly, and sand blast unprotected portions. The glued paper is easily removed by suspending the article in hot water. Subsequent processes, such as plating, etc., may then be proceeded with.

The Sand Blast for the Bath.—The last refinement of the sand blast is perhaps in connection with the bath, where it has been employed to give the last dainty touch to the human form divine, the operation being somewhat Frenchily sketched as follows.

"A few bushels of sand is brought to her room and after being slightly warmed it is spread out upon a sheet. A maid rubs her body all over with fine sandpaper, and after this process soft, rich cream is massaged into the skin. Then the bather stands in the middle of the sheet and taking up handfuls of the sand she rubs it over her body until she is glowing with the friction.

"Then she reclines at full length on the sheet, the ends of which are folded over her, and rests several moments before rolling over and over so as to become completely immersed in the sand. Then follows the 'blow-bath.' From a sort of fan-shaped blower sand is whirled out briskly so as to strike the body as forcibly as the bather can stand it. The effect is said to be wonderfully stimulating, and the sting is not unlike that produced by electricity."

Pneumatic Painting.—From the sand blast, conveying and forcibly discharging comminuted solids to impinge upon and abrade the surfaces with which it may come in contact, is an easy step to the conveyance and discharge of a true liquid instead of the sand, and then we have the various paint-spraying devices. These have in many cases been savers of labor where large surfaces were to be quickly and cheaply covered, their work being more satisfactory for whitewashing or kalsomining rather than for painting proper, although for the latter purpose pneumatic devices have been extensively used for painting freight cars and similar work. In this the results can scarcely be ranked very high, and are nothing for compressed air to boast of. As with the sand blast, however, pneumatic painting also has its dainty touches in some of the details of art work.

The Cement Gun.—An apparently important and a comparatively recent application of compressed air in the cement gun may be said to be a combination of the sand blast and the paint sprayer. The success or even the practicability of the cement gum could scarcely have been regarded with much confidence until demonstrated by actual experiment. There were two important questions which abstract thought could never have settled. The value and reliability of cement or concrete, assuming that the individual ingredients are what they should be, depends upon the maintenance of the correct proportions in the mixture and especially the quantity of water; and then as when the "gun" was used the cement was to be applied to surfaces at every angle from horizontal to vertical and from vertical to all the angles above the vertical to horizontal overhead.

Both questions are practically and most satisfactorily answered. In the handling of cement mortar in the ordinary way there is a theoretical proportion of water that is best for the material when set; but this theoretical amount of water forms a mortar too stiff to handle, while an excess of water weakens the mass by causing voids when the water has disappeared. In the cement-gun process it is claimed that no excess of water can remain in the mortar to later cause voids neither is it possible for the finest particles of the cement to begin to set before the mortar is placed. The cement gun appropriates only the necessary amount of water for the proper hydration of the cement, and as the materials are projected with considerable force all surplus water and air are expelled. The product is a non-porous, impermeable mass possessing the maximum density and practically waterproof.

As to the adhesion of the cement we have the following exposition from a paper by Mr. William A. Gordon in Transactions of American Society of Engineering Contractors:

If a sand blast is directed against a surface of a comparatively soft and sticky nature, such as ordinary wax, a very curious and unexpected thing happens. The particles of sand, instead of instantly tearing and wearing the wax away, as might be expected, penetrate the surface and stick, thus becoming a part of the mass itself. This process is continued until the wax is, as it were, saturated, and no more sand can find lodgment therein; in other words, the wax arms itself against attack with the sand itself, and thus the most powerful sand blast is rendered harmless.

Let us consider now what might be expected to happen if a certain proportion of powdered wax were introduced into the air blast with the sand, and the combined material directed against some hard surface, a sheet of boiler iron, for instance. Of course, the first particles of sand would strike the iron and bounce off, much as a pebble does when thrown against a wall; but, being soft and sticky, every atom of wax would adhere, and within a few moments the entire surface of the iron would be protected by a thin coating of wax. After this coating had become thick enough to enable the grains of sand to embed themselves therein, they would cease to bounce off as at first, but would stick and become a part of the mass or coating itself. It is easy to understand that, if the process were continued, the sand and wax could be made to form as thick a coating on the surface of the iron as might be desired.

If in the above experiment in place of wax we introduce cement and water, the operation of the cement gun is easily comprehended. When the nozzle is first directed against any hard surface, the particles of sand do not at first adhere; they fall away until a coating of cement sufficiently thick is formed to enable the sand to embed itself. When the cement gun is applying new cement to a body which has already set, the effect in practice is to deposit first a thin layer of practically neat cement where the new work joins the old, and after hardening if a test piece is broken it will be found that this joint or initial surface of adhesion is the strongest part of the mass.

The cement gun came, as we might say, from the outside, and was not first thought of, tried and brought to success by any engineer, builder or contractor. It was originally conceived by Mr. C. F. Akeley, a taxidermist of Chicago. His idea was to use the device to rapidly and economically build up the forms upon which to mount the skins of large animals, the promised advantage being that material could be added locally little by little to the limbs or bodies as judgment dictated, and it was a pronounced success at once.

It happened that Mr. Akeley was a member of the Field Museum committee, and had charge of the remodeling and preserving of one of the old World's Fair buildings in Jackson Park, Chicago, which had been given to the Field Museum Association. He produced an enlarged cement gun and employed it in covering the entire building with stucco work. The experiment was entirely successful, the work was done rapidly, well within the estimated cost, and the life work of the cement gun had commenced.

A brief description of the cement-gun apparatus and mode of operation is all that can be given here. It is generally necessarily portable, in the smaller installations the mixer carriage having an air-compressor and appurtenances upon it, making the arrangement entirely self-contained, while for work of larger and more extensive character the air supply is separate from the "gun" machine proper.

A mount of the latter type is sketched in Fig. 92, the air-supply coming from some other source and connected by suitable hose. A vertical section of the entire machine is shown with a reduced section of the mixing apparatus and also of a typical delivery nozzle.

The sand and cement, mixed in correct proportions and perfectly dry, is dumped into the upper chamber *A* through the swinging gate *C* which closes against a rubber gasket. Chamber *B* is separated from chamber *A* by a similar swinging gate *D* with its rubber gasket. The arrangement of these two chambers constitutes an air-lock which permits the intermittent insertion of material without interfering with the continuous operating of the gun. In the lower parts of *A* and *B* are vertically rotated agitators P, which are operated by hand. These are for breaking up any lumps of the mixture of cement and sand which might be overlooked by the charger.

The feed or discharge apparatus is ingenious and interesting. At the very bottom of B and below P is the main agitator or feed wheel L, which is operated by an air motor. This wheel is a solid steel plate about 1 in. thick with the edge notched all around, making it look like a cog wheel. The pockets in the edge formed by these serrations carry the dry mixture in rapid intermittent

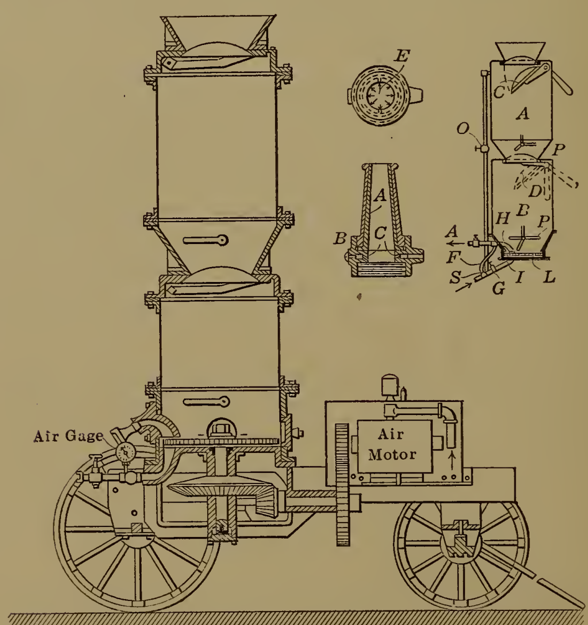

Fig. 92.—Cement Gun Apparatus.

charges over the "blow-pipe" I and its charge of compressed air, thus the mixture is blown upward and outward through the goose neck H into the flexible hose on the end of which is the nozzle. The driving air enters through the pipe S.

The gun is operated as follows: the gate C is open with gate D closed, at which time it is possible to run the air through B to test out the hose and nozzle, at the same time pouring a charge of the dry mixture in A. The gate C is now closed and the three-way valve O in the equalizing pipe opened, letting the compressed air into *A*. In a short time the air-pressure will hold *C* closed; a few seconds after this the pressure in *A* equals the pressure in *B*, and the gate *D*, which has been held in place by the air-pressure, opens automatically, due to the weight of the charge on the top and the equal air-pressure on both sides, permitting the charge in *A* to fall into *B*. Next the gate *D* is closed by hand and the air exhausted from *A*, when the gate *C* opens by its own weight. The air in *A* only has the duty of keeping *C* closed so that the pressure in *B* remains constant, assuring a uniform discharge from the feed hose. The gun is now ready for another charge of dry mixture.

The feed wheel *L* revolves on a center vertical shaft and is propelled by a small air driven-motor, direct-connected by bevel gears. As this wheel revolves, it carries a small amount of the dry mixture at a time directly into the main compressed-air current. *I* and *H* are hooded so that the current carries directly from the former to the latter, picking up the small charge of materials on the way. The output of the gun depends upon the speed of *L*, and this can be governed by a valve on the supply side of the air motor. Thus it is that the gun has a continuous feed, and is, therefore, continuous at the nozzle. To stop the stream of materials, simply shut *G* and the air to the motor. The bypass *F* and its valve is used to blow out the hose if it should clog. The air for shooting is admitted at *S* and on this line just beyond *G* is an air-pressure gage. The feed hose with the dry materials leads off from *R* to the nozzle.

The nozzle is of particular interest, as it is there the dry mixture first comes in contact with the water for hydration. The water under ordinary city pressure, or about 35 lb. per square inch, is delivered to the nozzle through a 1/2-in. flexible hose, which is entirely separate from the feed hose. At the larger end of the nozzle is a water sleeve *B*, completely around it, so that the water enters as a fine spray from eight 1/16-in. holes at *C*, from all sides impinging toward the center. The amount of water used is regulated by a valve at the nozzle. At first a great deal of trouble arose due to the wearing out, by the abrasive action of the sand and cement, of the smaller end of the nozzle, so now a rubber lining *A* is used, which holds up better than any substance yet tried.

The materials, in a wet state, leave the nozzle at about the rate of 300 ft. per second. From this fact the "gunite" is far denser than hand-applied, also of a greater tensile strength. A test which helps to bear this out was made. One ton of cement mortar applied by hand covered 25 sq. yd., 1 in. thick, while the gun covered only 16 1/2 sq. yd. 1 in. thick.

CHAPTER XXVIII

LIQUID AIR—OXYGEN FROM THE ATMOSPHERE

Air, like water, may exist either as a solid, as a liquid or as a vapor or gas. Water is most familiar to us in the liquid state, and this we think of as its normal condition. By artificial changes of temperature alone we convert it into a vapor. On account of association and familiarity we consider the normal state of the air to be the gaseous, but we have learned by changes of temperature and pressure, in the reverse direction, to convert it into a liquid or a solid.

The boiling-point of air, the liquid, at atmospheric pressure is $-312°$, or 524° below the atmospheric boiling-point of water. As with water, the boiling-point of liquid air varies with the pressure. At a pressure of 294 lb. the boiling-point is $-240°$ or 72° above the atmospheric boiling-point. At a pressure of 573 lb. the air changes from the liquid to the gaseous state, or vice versa, at a temperature of $-220°$. This is called the critical temperature of air, and no pressure, however great, can cause it to liquefy at a temperature higher than this.

For every vaporizable liquid there is a certain temperature and pressure at which it may be converted into the vaporous state in the same space which it occupies as a liquid, the temperature being the dominating condition, and when above this critical temperature a gas, whether a true gas, or a mixture of gases, which the air is, can be compressed down to the liquid volume of its mass without liquefying.

Liquid air is somewhat lighter than water, its specific gravity being 0.94. When liquid air is confined and alllowed to evaporate at ordinary atmospheric temperature it generates a pressure of about 12,000 lb. to the square inch. The relative volume of free air at atmospheric temperature and pressure as compared with it in its liquid state is about 800 to 1.

It must be confessed that cheap liquid air has been more or less of a disappointment to many engineers, and to the general public who have not been well informed as to its properties and possibilities. The impossibility of retaining it continuously in the liquid state was the first trouble; it boils away rapidly in spite of all that can be done to insulate it. Its low temperature at once suggested its employment for refrigerative purposes; but the commercially established refrigerative processes do the work so much more cheaply as to render the proposition almost an absurdity. Those who have tried to use the reëvaporating air for power purposes have had no better success, and the liquid air mine rescue apparatus has also demonstrated its impracticability.

In another direction of utilization, however, liquid air has already accomplished wonders, and its utilization in this direction is growing rapidly. This is as a source of oxygen supply for industrial and other uses, and in this field alone an enormous business has developed, the cheapness and readiness with which oxygen may now be supplied having led to the rapid growth of industrial processes of the highest practical value. The recent wonderful achievements in metal welding and cutting, as by the oxy-acetylene, oxy-benz and similar processes are all developments following the cheap production of liquid air.

After air is liquefied, if it then is allowed to reëvaporate the evaporation differs in an important particular from that of water. It is generally understood that the two principal gaseous constituents of water, oxygen and hydrogen, are united chemically, so that water may be evaporated and condensed back again, and the operation may be repeated over and over again and the water will continue to be water as at the beginning.

In the case of liquid air, however, this does not hold true. The two constituent gases, oxygen and nitrogen, do not so truly and constantly adhere to each other. The gases have different boiling-points, the nitrogen boiling away before the oxygen, so that there is provided a ready means of separating the two gases or of abstracting either of them individually and separately from the atmosphere to be used for any service that may develop for them.

Professor Carl von Linde succeeded in liquefying air in a commercially practical manner in 1895, and the possibility of the cheap production of oxygen was recognized almost at once. This has been developed into a great business already, the British Oxygen Company, employing the Linde processes, having eight large plants in different cities of Great Britain with an aggregate capacity of 400,000 cu. ft. of oxygen per day. The Linde Air

Products Company in the United States also has extensive plants in many cities, while Germany and other European nations are also active in the same direction.

Since 1895 there have of course been many improvements in methods and apparatus employed. In the earlier manipulations only about two-thirds of the oxygen could be abstracted from the air handled, but by the refinements of process developed only recently by M. Georges Claude nearly all the oxygen is secured. The entire matter has recently been presented to the public in a valuable unsigned article in *The Engineer*, London, April 4 and

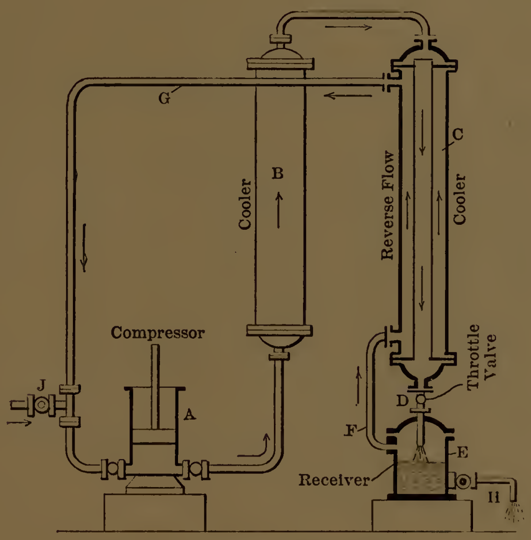

Fig. 93.—Linde Liquid-air Plant.

11, 1913, and what follows here is mostly abstracted from that article.

By the Linde methods the expensive laboratory processes for producing the low temperatures required were rendered obsolete, and a self-intensive procedure was substituted. Where a gas under pressure is allowed to expand through an orifice to a lower pressure without doing external work, the final temperature is slightly lower than the initial. For air initially at 63° the fall is 0.46° per atmosphere difference of pressure, so that if the expansion were from 11 atmospheres to 1 atmosphere the drop of temperature would be 4.6°. Linde used this apparently small cooling effect in a way to make it cumulative and so produced the low temperature required.

Air Liquefaction.—Fig. 93 is a diagram showing the essential features of the apparatus. Air compressed in cylinder *A* to, say, 75 atmospheres was passed through a water jacket cooler *B*, and its temperature was reduced to about 50°. On leaving this cooler the air traveled down the inner tube of a "reverse flow cooler" *C* and thence through a throttle valve *D* into a receiver *E*. With 75 atmospheres pressure above the throttle and 25 below it the free expansion of the air through the valve *D* lowered the temperature of the air by about 23°.

This cold air leaving the receiver passed by way of pipe *F* into the outer jacket of the reverse flow center. While flowing up this jacket it abstracted some of the heat from the succeeding charge of air flowing down the inner tube. If the interchange were completely effected the first charge of air would leave the reverse flow cooler by way of the pipe *G* at 50°, the temperature, that is, of the air leaving the water jacket cooler *B* . On the other hand, the second charge of air would reach the top side of the throttle valve at 27°, the temperature, that is, of the first charge of air *after* it had passed the throttle valve.

The first charge of air flowing through the pipe *G* was again compressed in the cylinder *A* and again cooled to 50° in the jacket cooler *B*. The second charge of air had by this time expanded through the throttle valve *D*, and its initial temperature of 27° had thereby become reduced about 29°, so that in the receiver its temperature would be −2°. The first charge was then cooled to −2° by the second charge in the reverse flow cooler, and the second charge again brought up to 50° and returned to the compressor. As this cycle was repeated each charge of air arrived at the throttle valve at a successively lower temperature until the critical temperature, −220°, was reached.

At this point the air, so far as temperature was concerned, was in a condition to be liquefied. The critical pressure for air at that temperature is, however, over 40 atmospheres, and, as the pressure in the receiver was only 25 atmospheres it is clear that even with the air at the critical temperature it would still be in the gaseous state. The cooling cycle was therefore carried still farther until the temperature of the air in the receiver was sufficiently low to enable the pressure of 25 atmospheres to produce liquefaction. For convenience the air has here been spoken of as in successive charges, but in practice the process was really continuous, and as liquid air was drawn off at *H*, fresh atmospheric air was introduced at *J*.

The Separation of the Oxygen.—Under ordinary atmospheric pressure the boiling-point of liquid nitrogen is −296.5°, and that of liquid oxygen is −320°, a difference of 23.5°, and it would seem a simple thing to boil away the nitrogen and retain the oxygen by keeping the liquid below its boiling-point; but it did not work out in that way.

It was quite practicable to boil off the nitrogen until the remaining liquid contained 60 per cent. of oxygen, but beyond

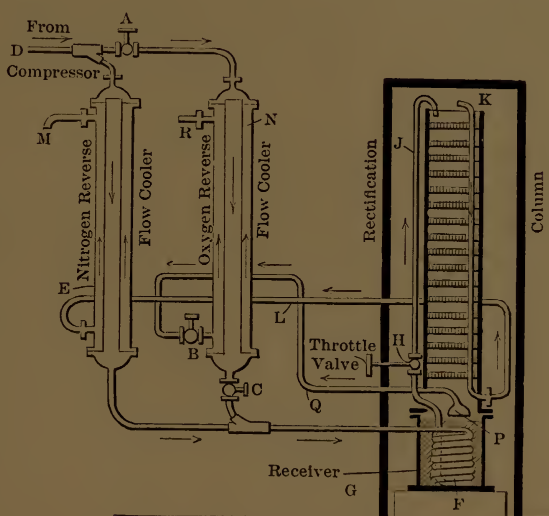

Fig. 94.—Linde Oxygen Plant.

that more and more oxygen was carried off with the nitrogen until the quantity of pure oxygen ultimately remaining was only a small fraction of the quantity present in the original volume of air. Not until 1902 could the rectification of liquid air be performed on a commercial basis. By Linde's method then devised he was able to extract no less than 93 per cent. of the oxygen from a given quantity of air, and to deliver it with a purity of about 99 per cent.

Referring now to Fig. 94, it is first to be supposed that the apparatus is just started up for a fresh run. Valves *A*, *B*, and *C* are closed, and air, compressed and cooled as before described, is entering the system at *D*. Flowing down the center pipe of the reverse flow cooler *E*, it passes through a worm *F* lying within a receiver *G*. Leaving the worm, the compressed air reaches the throttle valve *H*, where expanding to a lower pressure it is cooled thereby as before. The air thus cooled ascending pipe *J* is led to the top of the rectification column or chamber. Here it is sucked into pipe *K*, whence it is conducted through pipe *L* and the outer jacket of the reverse flow cooler *E* back to the compressor by way of pipe *M*.

With the exception of the worm *F*, the scheme is the same as in Fig. 93, so that after the apparatus has been working long enough the air leaving pipe *J* begins to carry with it an increasing quantity of liquid. Such liquid air, instead of passing over into pipe *K*, will fall down the rectification column and collect around worm *F* in receiver *G*.

We may suppose that the apparatus has been working long enough to have worm *F* completely immersed in liquid air. Valves *A*, *B* and *C* are now opened so as to bring a second reverse flow cooler *N* into parallel working with the first. The air from the compressor can thus flow through both coolers *E* and *N* on its way to the worm *F*. When it reaches this worm it is cooled by the surrounding liquid air, and on expanding through the throttle valve it emerges from pipe *J* in the liquid form.

As the compressor is now drawing direct from the atmosphere, a continuous stream of liquid air is falling down the rectifier from the pipe *J*. But if the air inside worm *F* is being cooled by the surrounding liquid air the latter must thereby be warmed. There is then passing down the rectifier liquid air, and passing upward is the cold gaseous air reëvaporated within the receiver *G*. The two streams are brought into intimate contact by means of perforated plate baffles arranged within the rectification column.

The vapors ascending the rectifier are at first very largely nitrogen. These nitrogenous vapors are at a sensibly higher temperature than the down coming current of liquid air, so that a transfer of heat takes place between the two currents. The comparatively hot nitrogenous vapors evaporate again some of the descending liquid air, this evaporation being accompanied by a predominance of nitrogen in the evaporate.

During these earlier stages of the working the original liquid air in the receiver *G* is from a two-fold cause becoming richer in oxygen and poorer in nitrogen. Ultimately the liquid surrounding the worm *F* will be practically pure oxygen, and when this stage is reached the apparatus begins to fulfil its function of producing oxygen, and incidentally nitrogen, as comparatively pure gases at ordinary atmospheric temperature.

The liquid air which flows out of pipe *J* in descending over the plates in the rectification column is brought into intimate contact with the ascending gaseous oxygen which itself has evaporated while passing through the worm. The cold liquid air and the slightly hotter gaseous oxygen have a transfer of heat, and the ascending gaseous oxygen is cooled and condensed and falls back into the receiver as liquid. The descending liquid air is heated; the more volatile constituent, nitrogen, is boiled off and ascends the column, while the oxygen, still liquid, continues its course downward to the receiver. The nitrogen gas is delivered into pipe *K*, travels along pipe *L*, and flows through the outer jacket of reverse cooler *E* to the atmosphere, or to a holder if it be desired to retain it.

The oxygen is drawn off by way of an inverted bell *P* fixed just above the surface of the liquid in receiver *G*, in which position it catches a portion of the oxygen vapors just after they are evaporated. The oxygen then flows by way of pipe *Q*, and the outer jacket of the reverse flow cooler *N*, through the pipe *R*, to a suitable gas holder. Within the coolers *E* and *N* respectively the nitrogen in pipe *L* and the oxygen in pipe *Q* are used to cool the incoming compressed air, so that by the time they flow away from the coolers at *M* and *R* they are practically at atmospheric temperature.

The above may be considered only a crude outline of the original Linde process as actually operated. There are many minor details not hinted at, and some features which it is still desirable to keep secret. A very important operation is the filtering and drying of the air at the beginning. Even with the most elaborate precautions all the moisture is not removed, and after a plant has been run about six days it "freezes up;" but as the plants are in duplicate, one can be running while the other is righting itself.

The Claude Process.—Now to understand the Claude process for producing oxygen from liquid air it is necessary to refer to Fig. 94 again to call attention to a fundamental factor of the process not so far explicitly mentioned.

When the apparatus is working steadily a constant stream of liquid air emerges from pipe J on to the top tray in the rectifying column. Simultaneously practically pure liquid oxgyen is present in the receiver G. The gas evaporating from this liquid and ascending the column is therefore also practically pure oxygen. At the foot of the column, then, the temperature is that corresponding to the boiling-point of liquid oxygen at atmospheric pressure, say, -296.5, and at the top the temperature is that of the liquid air, which may be taken as identical with the boiling-point of liquid nitrogen under atmospheric

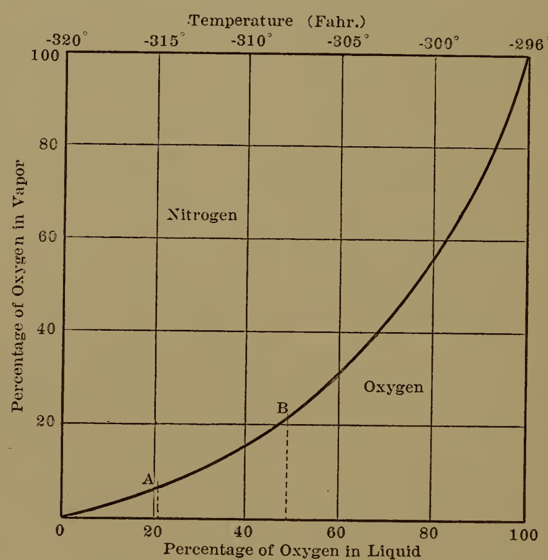

FIG. 95.—Diagram of Separation Process.

pressure, say, -320, and intermediate between the top and the bottom of the column the temperature lies between these limits. There is thus a temperature gradient set up in the rectifier, the lowest temperature being at the top and the highest temperature at the bottom. The iotablishment and maintenance of this temperature gradient ars essential features of the Linde rectification process, and must be considered a little more closely before the Claude improvements can be clearly understood.

In Fig. 95 a diagram is given graphically illustrating the fundamental points. Suppose that we have a closed vessel partially filled with a mixture of liquid nitrogen and liquid oxygen, and that the temperature of the vessel is somewhere between these boiling-points. The space above the level of the liquid will contain a vapor composed of a mixture of gaseous nitrogen and gaseous oxygen. We might not at first expect this. The temperature of the vessel being above the boiling-point of nitrogen, why should some of the nitrogen still be liquid? So also as the temperature of the vessel is below the boiling-point of oxygen, why should some of the oxygen be gaseous?

It may be that these apparently simple questions have never been satisfactorily answered, still it is clear that the quantity of liquid nitrogen which at the given temperature would be gaseous is in some way balancing a quantity of gaseous oxygen which otherwise would be liquid. What experiment tells us about the composition of the liquid and its vapor is embodied in the diagram, Fig. 95.

If we start with a liquid containing 100 per cent. of nitrogen, the vapor produced will be 100 per cent. nitrogen. So if the liquid contains 100 per cent. oxygen the vapor produced will also contain 100 per cent. oxygen. It is quite different, however, if our liquid is initially composed of n per cent. of oxygen and $100-n$ per cent. of nitrogen. The vapor driven off at first will contain m per cent. of oxygen and $100-m$ per cent. of nitrogen, and m in general will not be equal to n, but will be less than n. If $n=21$, the same percentage composition as the atmosphere, the vapor driven off at first will contain only 7 per cent. of oxygen, as shown by the point A on the diagram, Fig. 95.

Again, for m to be 21, or to obtain a vapor with the same composition as the atmosphere, we would require to start with a liquid containing 48 per cent. of oxygen, as designated by B on the diagram. The compositions of these liquids are somewhere between pure nitrogen on the one hand and pure oxygen on the other, and their boiling-points are between $-320°$ and $-296°$. The diagram shows the boiling-point for the liquid containing 21 per cent. of oxygen to be about $-315°$, and for that containing 48 per cent. about $-308°$.

If a liquid containing n per cent. of oxygen produced a vapor containing m per cent. of oxygen, the percentage composition of the liquid toward the end of the evaporation would be the same as at the beginning, provided m were equal to n. If, however, m is less than n the liquid will become increasingly rich in oxygen as the evaporation proceeds. If we start with a liquid containing 21 cu. ft. of oxygen and 79 cu. ft. of nitrogen, then according to the diagram, by the time we have evaporated 1 cu. ft. of the liquid we have removed from the liquid 0.07 cu. ft. of oxygen and 0.93 cu. ft. of nitrogen. Out of the 99 cu. ft. of liquid remaining 20.93 are oxygen and 78.07 are nitrogen. The percentage of oxygen in the liquid has risen from 21 to 21.14, and that of nitrogen has fallen from 79 to 78.86, and since the oxygen percentage has thus increased, the boiling-point will have risen to correspond. Hence, unless the temperature at which the evaporation is commenced is increased as time goes on, the liquid will cease to evaporate.

In Fig. 94, at the top of the column issuing from pipe J we have liquid air at a temperature of about $-320°$. As it falls down the column it experiences the influence of the temperature gradient and its temperature rises until it reaches the boiling-point of liquid air, $-315°$. Evaporation then commences and vapors are given off containing 7 per cent. of oxygen. The composition of the liquid begins to change; its percentage of oxygen increases and its boiling-point rises. The temperature gradient, however, also is rising, so that the descending stream of liquid is being continuously evaporated from the top to the bottom of the column.

Side by side with this process another is going on, namely, the condensation of the gaseous oxygen ascending from the receiver. How do these two processes affect one another, and what is the net composition of the vapor at any given point in the column? The mechanism of the interchange which goes on is difficult to state in concise and exact terms. The ultimate result is that at any given point in the rectification column the composition of the resultant vapor and the composition of the resultant liquid are functions of the temperature of the point, the relationship being as exhibited in Fig. 95. In other words, if a liquid containing n per cent. of oxygen and $100-n$ per cent. of nitrogen be intimately mixed with a vapor containing m per cent. of oxygen and $100-m$ per. cent of nitrogen, and with another vapor consisting of 100 per cent. oxygen, simultaneous condensation and evaporation will proceed until a resultant liquid and a resultant vapor are formed the compositions of which are as indicated in Fig. 95 under the temperature at which the mixing process is conducted.

When the steady state has been reached the vapor in the rectification column has at any given point a constant composition. It is pure oxygen at the foot; at the top is a mixture of 7 per cent. oxygen and 93 per cent. nitrogen and up and down the column is a regular graduation from one of these to the other. The vapor is therefore drawn off at the top of the column, for it is here that the oxygen percentage is least. The vapor drawn off into the pipe *K*, Fig. 94, is not pure nitrogen. It contains 7 per cent. of oxygen, or that percentage of oxygen which is found in the vapor arising from normal liquid air.

In the Linde process this loss of oxygen cannot be avoided. To obtain a less percentage of oxygen in the nitrogenous vapors drawn off through pipe *K* we would require something else than liquid air issuing from pipe *J*. We would in fact need to have a liquid richer in nitrogen and poorer in oxygen than is liquid air.

Broadly speaking, the Claude process consists in dividing the liquid air, before it reaches the rectification column, into two portions, one poorer in oxygen and the other richer in oxygen than the original liquid air. Each portion is introduced into the rectification column at the point on the temperature gradient corresponding to its respective boiling-point. The nitrogenous vapor carried off in the process still contains oxygen, but the percentage is less then in the Linde process, and, in fact, corresponds with the percentage contained in the evaporate of the liquid fraction poorest in oxygen.

Fig. 96 is a diagram of the Claude process drawn so as to be readily comparable with Fig. 94, the diagram of the Linde process. It will be noticed that, starting from the left, Fig. 96 is similar to Fig. 94 until the point *S* is reached. At this point a branch pipe passes upward and leads into a vessel called a liquefier. The main pipe at *S*, however, turns down and leads into an expansion engine, the exhaust from which is conducted to the rectification column. The by-pass going through the liquefier joins up to the exhaust pipe of the expansion engine through a throttle valve *I*. We first suppose that this valve is shut, so that the liquefier is out of action, also that valves *A*, *B*, and *C* are shut.

Under these circumstances, air from the compressor enters the system at *D* and passing down the reverse flow cooler *E*, reaches the point *S*. As the by-pass is, for the time being, a blind pipe, all the compressed air has to find its way through the expansion engine, which is coupled to a dynamo and made to do external work and the temperature of the exhaust is therefore low. The cold expanded air flows into a vessel *U*, from which two tubes pass upward into a vessel *V*, which in turn is conducted by two tubes passing downward with an annular vessel *W*. From *W* the cold air flows past the valve *X*, wide open at this time, up the pipe *J*, and into the interior of the rectifier. Part of the cold air leaves the vessel *U* by an alter-

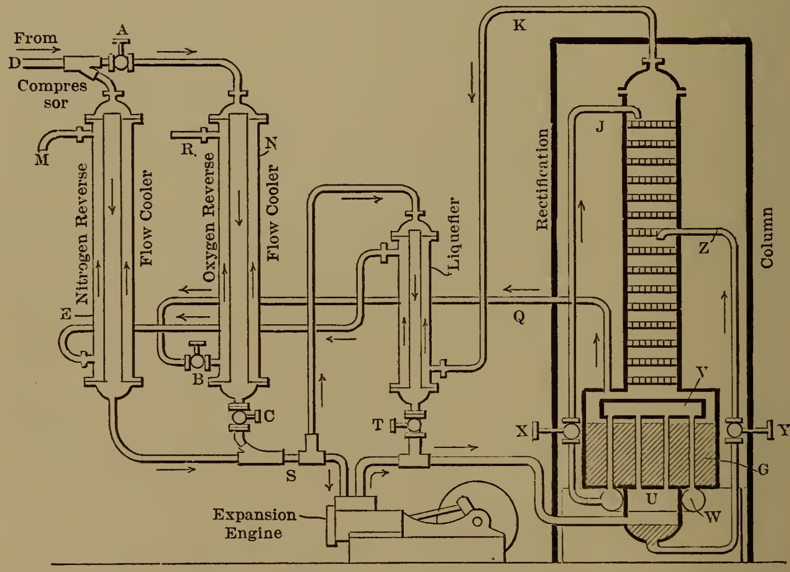

Fig. 96.—The Claude Oxygen Process.

native route consisting of a pipe *Z* leading from the foot of the vessel *U* through a wide open valve *Y* into the rectifier.

Both portions of cold air leave the rectifier by way of pipe *K*, and, flowing through the outer chamber of the liquefier, reach the reverse flow cooler *E*, whence they are conducted by the pipe *M* back to the compressor. The compressed air delivered to the expansion engine thus, as time goes on, becomes progressively cooler, and the expansion engine is fullfilling precisely the same duty as the throttle valve *H*, Fig. 94, of the Linde apparatus. It is also clear that during this preliminary stage of the operations the rectifying column, the vessels *U*, *V*, *W*, and the liquefier are all being cooled down by the cold exhaust of the expansion engine.

When the liquefier has been cooled sufficiently, valve *T* is opened and compressed air is by-passed from the point *S*. This air on issuing from the throttle valve *T* is converted into liquid, and as such is carried along the exhaust pipe by the exhaust from the engine into vessel *U*. It is not, however, allowed to remain here, but is swept up pipe *Z* by the exhaust into the interior of the rectifier. Falling downward it starts to collect in receiver *G* round the pipes leading into and out of vessel *V*. When sufficient liquid air has collected in the receiver, valves *A*, *B*, and *C* are opened and valves *X* and *Y* are partially closed so as to produce a throttling action.

Under these changed conditions the air reaching point *S* as before now divides, part going through the liquefier and part through the expansion engine. The two portions unite in the exhaust pipe and reach vessel *U* partly as liquid and partly as gas. This saturated vapor passes up the innermost pair of tubes into vessel *V*. In so doing it is cooled by the surrounding liquid air and suffers partial condensation. Since oxygen is more readily condensible than nitrogen it is clear that the liquid condensed at this stage will be richer in oxygen than is liquid air. Such rich liquid falls back into vessel *U*. The remaining gaseous portion of the air is naturally poorer in oxygen and richer in nitrogen by the result of this partial condensation. From vessel *V* it flows down the outer pair of tubes into the annular chamber *W*. Further condensation takes place, and as a result a liquid poor in oxygen collects in the chamber *W*.

Under the pressure of the exhaust of the expansion engine the liquids rich in oxygen and poor in oxygen are forced up pipes *Z* and *J*, respectively, past throttle valves *Y* and *X* into the rectifier. It will be noticed that pipe *J* enters the column at the top and pipe *Z* about halfway down. Inside the column the two portions of liquid come in contact with the vapors arising from the liquid air in receiver *G*. An exchange takes place, the ascending vapers growing richer in nitrogen and the descending liquids richer in oxygen. As time goes on the liquid in receiver *G* loses more and more nitrogen until, when the steady state is reached, it is pure, or practically pure, oxygen, just as in the case of the Linde apparatus previously described.

With the attainment of this stage a temperature gradient has been definitely established within the rectifier. At the foot there is the temperature corresponding to the boiling-point of liquid oxygen under atmospheric pressure. At the top there is the temperature corresponding to the boiling-point of the liquid ascending pipe *J*. As this liquid is poorer in oxygen than is liquid air, the temperature at the top of the Claude rectifier will be less than at the top of the Linde. Between the top and the bottom of the Claude rectifier there is one point on the temperature gradient which corresponds with the boiling-point of the liquid ascending pipe *Z*, the liquid, that is, which is rich in oxygen. Care is taken that pipe *Z* is located to enter the column just at this point.

In practice it is found that out of every three volumes of air, measured in the liquid state, entering vessel *U*, one volume passes up pipe *Z* as rich liquid, and two up pipe *J* as poor liquid. The composition of the rich liquid is roughly half oxygen and half nitrogen, while that of the pure liquid is about 6 per cent. oxygen and 94 per cent. nitrogen. The rectification process is precisely the same fundamentally in the Claude plant as in the Linde. At any point on the temperature gradient the composition of the vapor is in accordance with the data of Fig. 95. It follows, then, that at the top of the column, where the liquid contains only 6 per cent. of oxygen, the vapors driven off contain about 2 per cent. oxygen and 98 per cent. nitrogen. These nitrogenous vapors are carried off along pipe *K* and pass in succession through the outer jackets of the liquefier and the reverse flow cooler *E*, being finally allowed to escape. They contain 2 per cent. of oxygen, as compared with the 7 per cent. of the Linde process. The oxygen gas is drawn off from receiver *G* by way of pipe *Q*, and after being passed through the outer jacket of the reverse flow cooler *N* is conducted, by pipe *R*, to a suitable holder.

The purity of the gas produced by the Linde and the Claude processes is guaranteed as between 98.5 and 99.5 per cent. It is well known, however, that the purity is frequently as high as 99.8 per cent. It is claimed that the liquid-air process is the only means whereby oxygen entirely free from combustible residuals can be produced.

It is to be expected, perhaps, that some readers will not carefully go entirely through the preceding somewhat intricate description. Those who do cannot fail to realize at once its incompleteness. Many practical details of construction and arrangement, and of the means of manipulation and control are not even suggested. The sketches are necessarily misleading as to the relative and actual dimensions and capacities of the different elements, as for instance those of the compressor in Fig. 93 and of the expansion engine in Fig. 96. No information is given as to actual pressures and temperatures at the different points. One may well wonder and inquire how the man in charge of the apparatus can keep himself informed as to all that is going on throughout the series, even when everything is going well, and how he can discover when things are going wrong or determine what he should do to right them.

It is to be noted that nitrogen, the largest constituent of the atmosphere both in weight and bulk, is here represented as a discarded by-product, which it was completely in the beginning, but, like many other by-products, its value and the means and methods for its appropriation are being developed, which may ultimately mean not only nitrogen cheap and plenty, and a good demand for it, but in consequence still cheaper oxygen. With the clamor for nitrates for fertilizers the possibilities here opening cannot be ignored. By means of a simple evaporative device, several of which are already in use, it is possible without additional expenditure of power to eliminate the last traces of oxygen from part of the nitrogenous vapors coming away from the Linde and Claude rectifiers. In some cases, it is stated, plants have been established already for the production of nitrogen in the first instance with oxygen as the by-product.

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